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STANDARDS SUPPORT
AND
ENVIRONMENTAL IMPACT STATEMENT
VOLUME 1: PROPOSED STANDARDS OF PERFORMANCE FOR STATIONARY GAS TURBINES

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16. ABSTRACT

Standards of performance to control nitrogen oxides and sulfur dioxide emissions from new, reconstructed and modified stationary gas turbines in the U.S. are being proposed under section 111 of the Clean Air Act. This document contains information on the gas turbine industry and emission control technology, a discussion of the selected emission limitations and the supporting data and the alternatives which were considered, and analyses of the environmental and economic impacts of the proposed standards.

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Emission Standards and Engineering Division

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1. SUMMARY

1.1 PROPOSED STANDARDS

Standards of performance for stationary gas turbines are being proposed under section 111 of the Clean Air Act. These standards would limit emissions of nitrogen oxides and sulfur dioxide from stationary gas turbines whose peak load is equal to or greater than 10.7 gigajoules per hour heat input.

The numerical emission limit for NO_x would be 75 ppm by volume corrected to 15 percent oxygen and ISO ambient atmosphere conditions. The proposed standard would also include an adjustment factor for gas turbine efficiency and a fuel-bound nitrogen allowance. NO_x emissions from gas turbines, therefore, would be limited according to the following equation:

$$STD = (0.0075\ E) + F$$

where:

STD = allowable NO_x emissions (percent by volume at 15 percent oxygen)

E = efficiency adjustment factor:

$$\frac{14.4\ \text{kilojoules/watt·hr}}{\text{Actual ISO heat rate}}$$

F = fuel-bound nitrogen allowance:

Fuel-Bound Nitrogen (percent by weight)	F (NO_x - percent by volume)
$N \leq 0.015$	0
$0.015 < N \leq 0.1$	0.04(N)
$0.1 < N \leq 0.25$	0.004 + 0.0067 (N-0.1)
$N > 0.25$	0.005

During performance tests to determine compliance with the proposed standard, measured NO_x emissions at 15 percent oxygen would be adjusted to ISO ambient atmospheric conditions by the following correction factor:

$$NO_x = (NO_{x_{obs}})\left(\frac{P_{ref}}{P_{obs}}\right)^{0.5} e^{19(H_{obs}-0.00633)}$$

where:

NO_x = emissions of NO_x at 15 percent oxygen and ISO standard ambient conditions.

$NO_{x_{obs}}$ = measured NO_x emissions at 15 percent oxygen, ppmv.

P_{ref} = reference combustor inlet absolute pressure at 101.3 kilopascals (1 atmosphere) ambient pressure.

P_{obs} = measured combustor inlet absolute pressure.

H_{obs} = specific humidity of ambient air.

e = transcendental constant (2.718).

Gas turbine manufacturers, owners, or operators may develop their own factors for use during performance tests to adjust measured NO_x emissions to ISO ambient atmospheric conditions. These factors, however, must be in terms of the following variables: combustor inlet pressure, ambient air pressure, ambient air humidity and ambient air temperature. These factors must also be substantiated with data and approved by the Administrator for use.

The proposed standard for NO_x emissions would not apply to stationary gas turbines with a peak load heat input of less than 107.2 gigajoules per hour until five years following proposal. This would provide time for manufacturers to incorporate NO_x controls on these turbines.

Emergency gas turbines, fire-fighting gas turbines, military gas turbines and gas turbines creating ice fog would be exempt from the NO_x standard.

The numerical emission limit for SO_2 would be 150 ppm by volume corrected to 15 percent oxygen or a fuel sulfur content limit of 0.8 percent by weight. There would be no efficiency adjustment factor or ambient condition correction factor for SO_2 emissions, since SO_2 emissions are not affected by gas turbine efficiency or ambient atmospheric conditions. Also, no gas turbines would be exempt from the SO_2 emission limit or fuel sulfur content limit except those gas turbines whose peak load is less than 10.7 gigajoules per hour heat input.

The proposed standard requires continuous monitoring of the water-to-fuel ratio of a turbine using water or steam injection to control NO_x emissions and daily determination of the sulfur and nitrogen contents of the fuel fired by the gas turbine.

1.2 ENVIRONMENTAL/ECONOMIC IMPACT

Two alternative emission control systems were considered for selection as the basis for standards of performance limiting NO_x emissions from stationary gas turbines. These alternatives were dry controls, which consist mainly of different gas turbine combustor designs; and wet controls, which consist of water or steam injection into the gas turbine combustor. Dry controls would require approximately five years to incorporate on most gas turbines, whereas wet controls could be applied to large gas turbines now and could be applied to small gas turbines in about three years.

The main environmental benefit of a standard based on wet controls would be a reduction in NO_x emissions, totaling about 190,000 tons per year in 1981. Adverse environmental impacts associated with wet controls would be negligible. The impact on water and solid waste pollution would be small because the wastewater can be sewered directly. Energy consumption in the United States would increase only about 0.03 percent in 1981 due to wet controls on large gas turbines. There would be another 0.03 percent increase in energy consumption in 1986 due to application of wet controls on small gas turbines.

Standards based on dry controls would have no impact on national NO_x emissions in 1981 due to the necessity of allowing a five-year delay to incorporate dry controls on gas turbines. Dry controls on gas turbines would reduce NO_x emissions by about 90,000 tons per year in 1986. There would, however, be no water pollution, solid waste pollution, noise pollution or increased energy consumption associated with dry controls.

The economic impact associated with standards limiting NO_x emissions based on wet or dry controls would be approximately the same and would be small. The total annualized cost associated with the use of wet controls or dry controls would translate into price increases for the end products or services provided by gas turbine users of from 0.01 percent in the petroleum refining industry to 0.1 percent in the electric utility industry.

The environmental and economic impacts associated with these two control alternatives and the alternative of having no standard to limit NO_x emissions are summarized in Table 1. Based on this assessment of the impacts of standards of performance based on wet controls or dry controls, wet controls are selected as "...the best system of emission reduction (considering costs)..." for the reduction of NO_x emissions.

There are two possible control techniques for reducing SO_2 emissions from stationary gas turbines: flue gas desulfurization (FGD) and the firing of low sulfur fuels. FGD would cost about two to three times as much as the gas turbine. The economic impact of standards of performance based on FGD, therefore, is not considered reasonable.

Nearly all gas turbines currently fire low sulfur fuels and this situation is expected to continue at least through 1985. The impact on ambient air quality of standards of performance based on the firing of low sulfur premium distillate fuel oils, therefore, would be negligible. The economic impact would also be negligible and there would be no water, energy, solid waste or noise impacts associated with standards based on the firing of low sulfur fuel oils.

Based on this assessment of the impacts of standards of performance based on the firing of low sulfur fuel oils, this control technique is selected as "...the best system of emission reduction (considering costs)..." for the reduction of SO_2 emissions.

1.3 ECONOMIC IMPACT

An Economic Impact Analysis would be developed if the proposed standard caused an increase in operating costs in the fifth year of more than $100 MM, a major product price increase of 5 percent, or an increase in national energy consumption of 25,000 barrels of fuel oil per day. The proposed standard for stationary gas turbines would increase annualized costs by about 11 million dollars for large turbines by 1981 and by about 30 million dollars for all turbines by 1986. Energy consumption would increase by 5500 barrels of fuel oil per day for large turbines by 1981 and 7000 barrels per day for small turbines by 1986. The proposed standards would result in a price increase of 7.7 percent for gas turbines used on offshore platforms; however, a vast majority of these offshore turbines will have a peak load of less than 107.2 gigajoules per hour and will thus be exempt from the standard for five years. Some of the larger new offshore platforms will require turbines of peak load greater than 107.2

gigajoules per hour; however, these turbines will constitute such a small percentage of the overall market that the net price increase over the entire turbine market will be about 1 to 4 percent. The Agency, therefore, feels that no Economic Impact Analysis is required.

Table 1 - Environmental and Economic Impacts of Alternative Standards

Action \ Impact	Air	Water	Solid Waste	Noise	Energy	Economic	Inflation
Wet Controls (Proposed Standards)	+3	-1	-1	0	-1	-1	-1
Dry Controls	+2	0	0	0	0	-1	-1
No Action	0	0	0	0	0	0	0

Key
+ Beneficial Impact
- Adverse Impact

0 No Impact
1 Negligible Impact
2 Small Impact
3 Moderate Impact
4 Large Impact

2. INTRODUCTION

Standards of performance under section 111 of the Clean Air Act are proposed following a detailed investigation of air pollution control methods available to the affected industry and the impact of their costs on the industry. This document summarizes the information obtained from such a study of gas turbines. Its purpose is to explain in detail the background and basis of the proposed standards and to facilitate analysis of the proposed standards by interested persons, including those who may not be familiar with the many technical aspects of the industry. To obtain additional copies of this document or the Federal Register notice of proposed standards, write to Public Information Center (PM-215), Environmental Protection Agency, Washington, D.C. 20460 (specify "Standards Support and Environmental Impact Statement - An Investigation of the Best Systems of Emission Reduction for Stationary Gas Turbines").

2.1 AUTHORITY FOR THE STANDARDS

Standards of performance for new stationary sources are developed under section 111 of the Clean Air Act (42 U.S.C. 1857c-6), as amended in 1970. Section 111 requires the establishment of standards of performance for new stationary sources of air pollution which "...may contribute significantly to air pollution which causes or contributes to the endangerment of public health or welfare." The Act requires that standards of performance for such sources reflect "...the degree of emission limitation achievable through the application of the best system of emission reduction which (taking into account the cost of achieving such reduction) the Administrator determines has been adequately demonstrated." The standards apply only to stationary sources, the construction or modification of which commences after regulations are proposed by publication in the Federal Register.

Section 111 prescribes three steps to follow in establishing standards of performance.

1. The Administrator must identify those categories of stationary sources for which standards of performance will ultimately be promulgated by listing them in the Federal Register.
2. The regulations applicable to a category so listed must be proposed by publication in the Federal Register within 120 days of its listing. This proposal provides interested persons an opportunity for comment.
3. Within 90 days after the proposal, the Administrator must promulgate standards with any alterations he deems appropriate.

Standards of performance, by themselves, do not guarantee protection of health or welfare; that is, they are not designed to achieve any specific air quality levels. Rather, they are designed to reflect best demonstrated technology (taking into account costs) for the affected sources. The overriding purpose of the collective body of standards is to maintain existing air quality and to prevent new pollution problems from developing.

Previous legal challenges to standards of performance have resulted in several court decisions[1,2] of importance in developing future standards. In cases the principal issues were whether EPA: (1) made reasoned decisions and fully explained the basis of the standards, (2) made available to interested parties the information on which the standards were based, and (3) adequately considered significant comments from interested parties.

Among other things, the court decisions established: (1) that preparation of environmental impact statements is not necessary for standards developed under section 111 of the Clean Air Act because, under that section, EPA must consider any counter-productive environmental effects of a standard in determining what system of control is "best;" (2) in considering costs it is not necessary to provide a cost-benefit analysis; (3) EPA is not required to justify standards that require different levels of control in different industries unless such different standards may be unfairly discriminatory; and (4) it is sufficient for EPA to show that a standard can be achieved rather than that it has been achieved by existing sources.

Promulgation of standards of performance does not prevent State or local agencies from adopting more stringent emission limitations for the same sources. On the contrary, section 116 of the Act (42 U.S.C. 1857-D-1) makes clear that States and other political subdivisions may enact more restrictive standards. Furthermore, for heavily polluted areas, more stringent standards may be required under section 110 of the Act (42 U.S.C. 1857c-5) in order to attain or maintain national ambient air quality standards prescribed under section 109 (42 U.S.C. 1857c-4). Finally, section 116 makes clear that a State may not adopt or enforce less stringent new source standards than those adopted by EPA under section 111.

Although standards of performance are normally structured in terms of numerical emission limits where feasible,[1] alternative approaches are sometimes necessary. In some cases physical measurement of emissions from a new source may be impractical or exorbitantly expensive. For example, emissions of hydrocarbons from storage vessels for petroleum liquids are greatest during storage and tank filling. The nature of the emissions (high concentrations for short periods during filling and low concentrations for longer periods during storage) and the configuration of storage tanks make direct emission measurement impractical. Therefore, a more practical approach to standards of performance for storage vessels has been equipment specifications.

2.2 SELECTION OF CATEGORIES OF STATIONARY SOURCES

Section 111 directs the Administrator to publish and from time to time revise a list of categories of sources for which standards of performance are to be proposed. A category is to be selected "...if [the Administrator] determines it may contribute significantly to air pollution which causes or contributes to the endangerment of public health or welfare."

Considerable attention has been given to the development of a system for assigning priorities to various source categories. In brief, the approach that has evolved is as follows. Specific areas of interest are identified by considering the broad strategy of the Agency for implementing the Clean Air Act. Often, these "areas" are actually pollutants which are primarily emitted by stationary sources. Source categories which emit these pollutants are then evaluated and ranked by a process involving such factors as (1) the level of emission control (if any) already required by State regulations; (2) estimated levels of control that might result from standards of performance for the source category; (3) projections of growth and replacement of existing facilities for the source category; and (4) the estimated incremental amount of air pollution that could be prevented, in a pre-selected future year, by standards of performance for the source category. An estimate is then made of the time required to develop a standard. In some cases, it may not be feasible to develop a standard immediately for a source category with a high priority. This might occur because a program of research and development is needed to develop control techniques or because techniques for sampling and measuring emissions may require refinement.

Selection of the source category leads to another major decision: determination of the types of facilities within the source category to which the standard will apply. A source category often has several facilities that cause air pollution. Emissions from some of these facilities may be insignificant or very expensive to control. An investigation of economics may show that, within the costs that an owner could reasonably afford, air pollution control is better served by applying standards to the more severe pollution problems. For this reason (or perhaps because there may be no adequately demonstrated system for controlling emissions from certain facilities), standards often do not apply to all sources within a category. For similar reasons, the standards may not apply to all air pollutants emitted by such sources. Consequently, although a source category may be selected to be covered by a standard of performance, not all pollutants or facilities within that source category may be covered by the standards.

2.3 PROCEDURE FOR DEVELOPMENT OF STANDARDS OF PERFORMANCE

Congress mandated that sources regulated under section 111 of the Clean Air Act be required to utilize the best system of air pollution control (considering costs) that has been adequately demonstrated at the time of their design and construction. In so doing, Congress sought to:

1. Maintain existing high-quality air,
2. Prevent new air pollution problems, and
3. Ensure uniform national standards for new facilities.

Standards of performance, therefore, must (1) realistically reflect best demonstrated control practice; (2) adequately consider the cost of such control; (3) be applicable to existing sources that are modified as well as new installations; and (4) meet these conditions for all variations of operating conditions being considered anywhere in the country.

The objective of a program for development of standards is to identify the best system of emission reduction which "has been adequately demonstrated (considering cost)." The legislative history of section 111 and the court decisions referred to earlier make clear that the Administrator's judgment of what is adequately demonstrated is not limited to systems that are in actual routine use. Consequently, the investigation may include a technical assessment of control systems which have been adequately demonstrated but for which there is limited operational experience. In most cases, determination of the "degree of emission limitation achievable" is based on results of tests of emissions from existing sources. This has required worldwide investigation and measurement of emissions from control systems. Other countries with heavily populated, industrialized areas have sometimes developed more effective systems of control than those used in the United States.

Since the best demonstrated systems of emission reduction may not be in widespread use, the data base upon which standards are developed may be somewhat limited. Test data on existing well-controlled sources are obvious starting points in developing emission limits for new sources. However, since the control of existing sources generally represents retrofit technology or was originally designed to meet an existing State or local regulation, new sources may be able to meet more stringent emission standards. Accordingly, other information must be considered and judgment is necessarily involved in setting proposed standards.

A process for the development of a standard has evolved. In general, it follows the guidelines below.

1. Emissions from existing well-controlled sources are measured.
2. Data on emissions from such sources are assessed with consideration of such factors as: (a) the representativeness of the source tested (feedstock, operation, size, age, etc.); (b) the age and maintenance of the control equipment tested (and possible degradation in the efficiency of control of similar new equipment even with good maintenance procedures); (c) the design uncertainties for the type of control equipment being considered; and (d) the degree of uncertainty that new sources will be able to achieve similar levels of control.
3. During development of the standards, information from pilot and prototype installations, guarantees by vendors of control equipment, contracted (but not yet constructed) projects, foreign technology, and published literature are considered, especially for sources where "emerging" technology appears significant.
4. Where possible, standards are developed which permit the use of more than one control technique or licensed process.
5. Where possible, standards are developed to encourage (or at least permit) the use of process modifications or new

processes as a method of control rather than "add-on" systems of air pollution control.

6. Where possible, standards are developed to permit use of systems capable of controlling both gaseous and particulate matter emissions.

7. Where appropriate, standards for visible emissions are developed in conjunction with concentration/mass emission standards. The opacity standard is established at a level which will require proper operation and maintenance of the emission control system installed to meet the concentration/mass standard on a day-to-day basis, but not require the installation of a control system more efficient or expensive than that required by the concentration/mass standard. In some cases, however, it is not possible to develop concentration/mass standards, such as with fugitive sources of emissions. In these cases, only opacity standards may be developed to limit emissions.

2.4 CONSIDERATION OF COSTS

Section 111 of the Clean Air Act requires that cost be considered in developing standards of performance. This requires an assessment of the possible economic effects of implementing various levels of control technology in new plants within a given industry. The first step in this analysis requires the generation of estimates of installed capital costs and annual operating costs for various demonstrated control systems, each control system alternative having a different overall control capability. The final step in the analysis is to determine the economic impact of the various control alternatives upon a new plant in the industry. The fundamental question to be addressed is whether or not a new plant would be constructed if a certain level of control costs would be incurred. Other issues that are analyzed are the effects of control costs upon product prices and product supplies, and producer profitability.

The economic impact upon an industry of a proposed standard is usually addressed both in absolute terms and by comparison with the control costs that would be incurred as a result of compliance with typical existing State control regulations. This incremental approach is taken since a new plant would be required to comply with State regulations in the absence of a Federal standard of performance. This approach requires a detailed analysis of the impact upon the industry resulting from the cost differential that exists between a standard of performance and the typical State standard.

The costs for control of air pollutants are not the only costs considered. Total environmental costs for control of water pollutants as well as air pollutants are analyzed wherever possible.

A thorough study of the profitability and price-setting mechanisms of the industry is essential to the analysis so that an accurate estimate of potential adverse economic impacts can be made. It is also essential to know the capital requirements placed on plants in the absence of Federal standards of performance so that the additional capital requirements necessitated by these standards can be placed in the proper perspective. Finally, it is necessary to recognize any constraints on capital availability within an industry as this factor also influences the ability of new plants to generate the capital required for installation of the additional control equipment needed to meet the standards of performance.

A consideration of the impact of these standards on inflation is of major importance. Any action which will add significantly to inflationary pressures is considered major and requires an inflationary impact statement.

2.5 CONSIDERATION OF ENVIRONMENTAL IMPACTS

Section 102(2)(c) of the National Environmental Policy Act (NEPA) of 1969 (PL 91-190) requires Federal agencies to prepare detailed environmental statements on proposals for legislation and other major Federal actions significantly affecting the quality of the human environment. The objective of NEPA is to build into the decision-making process of Federal agencies a careful consideration of all environmental aspects of proposed actions.

As mentioned earlier, in a number of legal challenges to standards of performance for various industries, the Federal Courts of Appeals have held that environmental impact statements need not be prepared by the Agency for proposed actions under section 111 of the Clean Air Act. Essentially, the Federal Courts of Appeals have determined that "...Section 111 of the Clean Air Act, properly construed, requires the functional equivalent of a NEPA impact statement" in the sense that the criteria "...the best system of emission reduction," "...require(s) the Administrator to take into account counter-productive environmental effects of a proposed standard, as well as economic costs to the industry..." On this basis, therefore, the Courts "...established a narrow exemption from NEPA for EPA determinations under section 111."[1,2]

In addition to these judicial determinations, the Energy Supply and Environmental Coordination Act (ESECA) of 1974 (PL-93-319) specifically exempted proposed actions under the Clean Air Act from NEPA requirements. According to section 7(c)(1), "No action taken under the Clean Air Act shall be deemed a major Federal action significantly affecting the quality of the human environment within the meaning of the National Environmental Policy Act of 1969."

The Agency has concluded, however, that the preparation of environmental impact statements could have beneficial effects on certain regulatory actions. Consequently, while not legally required to do so by section 102(c)(2) of NEPA, environmental impact statements will be prepared for various regulatory actions, including standards of performance developed under section 111 of the Clean Air Act. This voluntary preparation of environmental impact statements, however, in no way legally subjects the Agency to NEPA requirements.

To implement this policy, therefore, a separate section is included in this document which is devoted solely to an analysis of the potential environmental impacts associated with the proposed standards. Both adverse and beneficial impacts in such areas as air and water pollution, increased solid waste disposal, and increased energy consumption are identified and discussed. Appendix B of this document outlines those sections or chapters which examine these potential environmental impacts in detail.

2.6 IMPACT ON EXISTING STANDARDS

Standards of performance may affect an existing source in either of two ways. Section 111 of the Act defines a new source as "any stationary source, the construction or modification of which is commenced after the regulations are proposed." Consequently, if an existing source is modified after proposal of the standards, with a subsequent increase in air pollution, it is subject to standards of performance. [Amendments to the general provisions of Subpart A of 40 CFR Part 60 to clarify the meaning of the term modification were promulgated in the Federal Register on December 16, 1975 (40 CFR 58416).]

Secondly, promulgation of a standard of performance requires States to establish standards of performance for existing sources in the same industry under section 111(d) of the Act if the standard for new sources limits emissions of a pollutant for which air quality criteria have not been issued under section 108 or which has not been listed as a hazardous pollutant under section 112. If a State does not act, EPA must establish such standards. [General provisions outlining procedures for control of existing sources under section 111(d) were promulgated on November 17, 1975 as Subpart B of 40 CFR Part 60 (40 CFR 53340).]

2.7 REVISION OF STANDARDS OF PERFORMANCE

Congress was aware that the level of air pollution control achievable by any industry may improve with technological advances. Accordingly, section 111 of the Act provides that the Administrator may revise such standards from time to time. Although standards proposed and promulgated by EPA under section 111 are designed to require installation of the "...best system of emission reduction...(taking into account the cost) ..."the standards will be reviewed periodically. Revisions will be proposed and promulgated as necessary to assure that the standards continue to reflect the best systems that become available in the future. Such revisions will not be retroactive but will apply to stationary sources constructed or modified after proposal of the revised standards.

2.8 REFERENCES

1. Portland Cement Association vs. Ruckelshaus, 486 F. 2nd 375 (D.C. Cir. 1973).

2. Essex Chemical Corp. vs. Ruckelshaus, 486 F. 2nd 427 (D.C. Cir. 1973).

3. THE STATIONARY GAS TURBINE INDUSTRY AND PROCESS

3.1 GENERAL

3.1.1 Introduction

A gas turbine is a rotary engine, of which a common example is the aircraft jet engine. The hot combustion gases from a stationary gas turbine drive a power-output shaft rather than providing thrust as on the aircraft engine. Figure 3-1 presents a cut-away view of a gas turbine. Turbines range in size from less than 40 horsepower (0.03 megawatts (MW)) to over one hundred thousand horsepower (75 MW). Manufacturers continue to increase the horsepower of individual turbines and frequently turbines are "ganged" or installed in groups, so that the combined power output from one location may exceed 1,648,000 horsepower (1230 MW).[1]

The use of stationary gas turbines has increased tremendously since the mid-1960's. Several characteristics of gas turbines which contributed to this growth are:

* can be instrumented for remote operation and, therefore, can be installed almost anywhere;

* Time from placement of an order to on-line operation of a turbine is relatively short;

* quick and easy installation;

* high horsepower to size ratio;

* short start-up time;

* relatively vibration free operation;

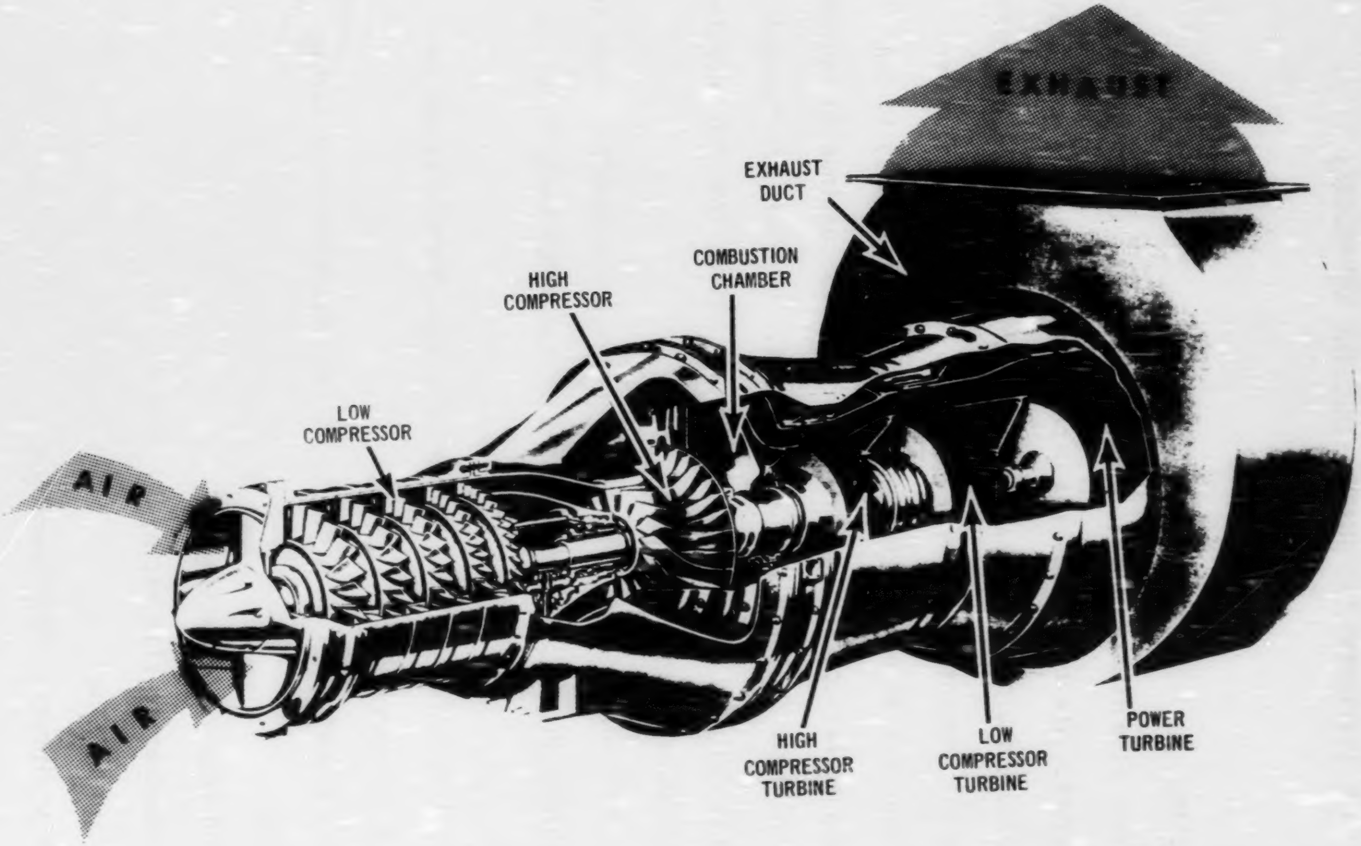
EXHAUST
EXHAUST DUCT
COMBUSTION CHAMBER
HIGH COMPRESSOR
LOW COMPRESSOR
AIR
AIR
HIGH COMPRESSOR TURBINE
LOW COMPRESSOR TURBINE
POWER TURBINE

Figure 3-1. Cut-away view of a typical stationary gas turbine.

- good reliability;
- capability of operation with a variety of fuels in any environment;
- operate without cooling water;
- have lower capital costs than diesel engines;
- have a very low physical profile (low buildings, short stacks, little visible emissions, and quiet operation when properly muffled).

3.1.2 Gas Turbine Industry Profile[2] 1/

This section discusses the basic structure of the gas turbine industry. It contains analyses of the various market sectors and growth projections showing the type and number of units likely to be affected by the new source performance standards.

The analysis is divided into five sections:

(1) description of turbine manufacturers;
(2) market description;
(3) historical market trends;
(4) market projections by application; and
(5) effect on the balance of trade position.

The analysis uses 1974 data for the thirteen manufacturers and packagers of turbines. All information presented, unless otherwise indicated, is in terms of orders placed rather than equipment delivered or installed.

3.1.2.1 The Manufacturers and Major Sellers of Gas Turbines - The United States stationary gas turbine industry consists of thirteen firms. Of those firms, only eight actually produce their own gas turbines. Table 3-1 lists the eight manufacturers and five packagers of stationary gas turbines. Cooper-Bessemer,

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1/ Unless otherwise noted, Reference 2 is the source of all market information in Section 3.1.2 and its subsections. Specific references for statements in Section 3.1.2 are contained in Reference 2.

Table 3-1. AMERICAN FIRMS MARKETING STATIONARY GAS TURBINES

Company	Parent Firm	Fortune 500 Rank of Parent Firms	Equipment Size Range 2/
AiResearch Manufacturing Co., Phoenix, Arizona	Garrett Corporation/Signal Companies, Inc.	121	680 HP - 800 HP
Avco Lycoming Division, Stratford, Connecticut	Avco Company	283	500 HP - 5673 HP
Cooper-Bessemer Company 1/ Mt Vernon, Ohio	Cooper Industries, Inc.	407	3050 HP - 36400 HP
Curtiss-Wright Corporation 1/ Woodridge, New Jersey	Curtiss-Wright	512	8150 HP - 398 MW
Detroit Diesel Allison Division, Indianapolis, Indiana	General Motors Corporation	2	260 HP - 4365 HP
Dresser-Clark Division 1/ Houston, Texas	Dresser Industries, Inc.	146	1500 HP - 26600 HP
Energy Transformation Corp., Bayertown, Pennsylvania	--	--	15 KW - 40 KW
General Electric Company Schenectady, New York	General Electric Company	8	4900 HP - 364.5 MW
Ingersoll-Rand Company 1/ Phillipsburg, New Jersey	Ingersoll-Rand Company	142	4250 HP - 24000 HP
Solar Division San Diego, California	International Harvester Company	26	40 HP - 3830 HP
Turbodyne Corporation 1/ St Cloud, Minnesota	Studebaker-Worthington, Inc.	150	65 MW - 430 MW
Turbo Power and Marine Systems, Inc. Farmington, Connecticut	United Aircraft Corporation	46	36900 HP - 350 MW
Westinghouse Electric Corporation Lester, Pennsylvania	Westinghouse Electric Corporation	19	19 MW - 636 MW

1/ Packager of turbine driven equipment.

Sources: Adapted from Gas Trubine Computer Marketing Services, Inc., data base.
The Fortune Double 500 Directory
Sawyer's Gas Turbine Catalog - 1975 Edition

2/ This "equipment size" treats multiple turbines as used in a combined cycle as one piece of equipment.

Curtiss-Wright, Dresser Industries, Turbodyne, and Ingersoll-Rand package and sell turbine units that they purchase from some other company. The English firm, Rolls-Royce, is the major supplier of gas turbines to four of the firms while Turbodyne packages Brown Boviere turbines.

The industry is dominated by giants. As shown in Table 3-1, seven of the eight manufacturers are owned by companies in the Fortune 500 rankings; and five of the eight are among the 50 largest manufacturing firms in the United States. The table also provides information on the size range of the firms' currently available equipment. The wide range of sizes indicates that some of the firms compete in different markets.

Table 3-2 ranks the turbine manufacturers and packagers by 1974 horsepower output. Because turbines vary widely in size, horsepower provides a better surrogate for dollar volume sales than number of units sold. The industry is very concentrated with the three largest companies holding over 80 percent of the market. The corporations manufacturing stationary gas turbines are all highly diversified, and six of these also produce reciprocating (i.e., diesel) engines, the gas turbine's major competitor. It is interesting to note that the top three gas turbine manufacturers (General Electric, Turbo Power and Marine, and Westinghouse) do not produce reciprocating engines as part of their overall company organization. Any regulatory change, therefore, that seems to favor reciprocating engines relative to turbines might be most strongly opposed by these companies.

The market for gas turbines is worldwide. In 1974, approximately half of all unit sales and 30 percent of all horsepower manufactured was sold outside the United States. While the regulations under consideration apply only to turbines sold in the United States, changes in the manufacturing process could also increase the price of turbine exports.

Table 3-2. STATIONARY GAS TURBINE FIRMS RANKED BY TOTAL 1974 ORDERS

Firm	1974 HP (000)	1974 % Market Share of Total HP	1974 Units	Parent Corporation Manufacturers Reciprocating Engines
General Electric	6,243.7	34.3	104	No
TP & M	5,204.3	28.6	40	No
Westinghouse	3,626.0	19.9	34	No
Turbodyne[1/]	580.4	5.1	7	Yes
Solar[3/]	880.9	4.8	509	Yes
Detroit Diesel Allison	NA	NA	NA	Yes
Avco-Lycoming	223.8	1.3	84	No
Curtiss-Wright[2/]	201.7	1.1	4	No
Cooper-Bessemer[2/]	163.6	.9	11	Yes
Ingersoll-Rand[2/]	152.7	.8	26	Yes
Dresser-Clark[2/]	88.5	.5	6	Yes
AiResearch/Garrett	49.7	.3	72	No
Energy Transformation	.1	--	2	No
TOTALS	17,415.4	100	899	

[1/] Company sales data used for all figures.

[2/] Packager of turbine driven equipment

[3/] Company sales data used for 350 HP models (1974 = date of shipment)

Source: Adopted from Gas Turbine Computer Marketing Service, Inc., data base.

3.1.2.2 The Market for Turbines by Equipment Size Range - Gas turbines have an extremely broad size range that starts under 100 HP and approaches 100 megawatts (134,000 HP). Generally, units over 20,000 HP are used for electric power generation and units under 20,000 HP are used for a variety of purposes including gas compression, stand-by power and miscellaneous industrial uses.

Table 3-3 shows the 1974 stationary gas turbine market divided by equipment size. Over 70 percent of the units sold in the United States are less than 5,000 HP while over 85 percent of the HP sold is in unit sizes of 50 MW or greater. The mid-market area from 5,000 HP to 50 MW contains less than 10 percent of the horsepower and units sold in the United States. This market area, however, is more significant outside the United States, accounting for over 20 percent of horsepower sales worldwide.

Table 3-4 presents the firms competing in each size range showing 1974 domestic orders by units and horsepower. The table indicates that there are two competitive groupings. The smaller firms compete in the market for units less than 5,000 HP while the four largest firms dominate the remainder of the domestic market. Only one packager, Curtiss-Wright, competes in the large horsepower end of the market.

The size categories can be divided into four groups as shown below:

Turbine Sizes and Applications

Size Category	Equipment Size	Number of Firms Competing	
		Mfg.	Pkg.
Small Capacity	5,000 HP	2	1
Medium Capacity	5,000-20,000 HP	2	3
Large Capacity	20,000-134,000 HP	3	2
Multiple and Combined Cycle[1]	134,000 HP	2	0

These groups are unique in terms of competition or types of applications.

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[1] See Section 3.2.1.3 for definition of a combined cycle turbine.

Table 3-3.

BREAKDOWN OF TOTAL 1974 STATIONARY GAS TURBINE ORDERS AND SHARE OF MARKET FOR SIZE RANGES, BY UNITS AND CAPACITY[1/]

	Total Orders				Share of Market (%)			
	Units		HP(000)		Units		HP	
HP/MW Range	U.S.	World	U.S.	World	U.S.	World	U.S.	World
0 - 1000 HP	178	209	77.0	97.0	9.9	8.9	.2	.2
1001 - 3000 HP	210	312	297.9	500.8	48.4	38.7	2.4	2.8
3001 - 5000 HP	59	168	228.0	644.6	13.6	20.8	1.8	3.6
5001 - 10000 HP	1	2	8.8	13.8	.2	.2	.1	.1
10001 - 20000 HP	11	32	167.1	1048.4	2.5	8.9	1.4	5.9
15 - 25 MW 20115 - 33526 HP	6	51	196.6	1577.3	1.4	6.3	1.6	8.5
25.1 - 50 MW 33526 - 67051 HP	13	22	498.6	867.4	3.9	3.2	6.2	6.7
50.1 - 100 MW 67,052 - 132,102 HP	66	80	5767.0	6965.7	15.5	9.9	48.4	39.8
100 MW 134,103 HP	20	23	4623.6	5700.4	4.6	2.8	37.7	32.2
TOTAL	564	899	11864.6	17415.4	100.0	100.0	100.0	100.0

1/ Less military applications.

SOURCE: Adapted from Gas Turbine Computer Marketing Service, Inc. data base.

Table 3-4.

BREAKDOWN OF FIRM'S 1974 ORDERS FOR STATIONARY GAS TURBINES BY SIZE RANGE, WITH UNITS AND CAPACITY

Firm	1-1000 HP		1001-3000 HP		3001-5000 HP		5001-10000 HP		10001-20000 HP		15-25 MW 20115-33526 HP		25.1-50 MW 33527-67051 HP		50.1-100 MW 67052-132102 HP		100 MW 134.103 HP	
	Units	HP	Units	HP	Units	HF	Units	HP	Units	HP	Units	HP	Units	HP	Units	HP	Units	HP
General Electric							1	8.8	8	107.1	6	196.6	4	103.4	40	3509.0		
TP & M									3	60.0			3	118.1	12	903.4	17	3992.0
Westing-house													3	145.0	8	870.2	3	731.6
Turbodyne[1/]															6	484.4		
Solar[2/]	135	47.3	184	226.5	45	172.6												
Lycoming			26	71.4	1	3.4												
Detroit Diesel Allison					NA													
AiResearch/ Garrett	43	29.7																
Others (Major Packagers)					13	52.0							3	132.1				
TOTALS	178	77.0	210	297.9	59	228.0	1	8.8	11	167.1	6	196.6	13	498.6	66	5767.0	20	4623.6

Note: HP = (000)

1/ Company sales data used for all figures

2/ Company sales data used for 350 HP model only (1974 = date of shipment).

3.1.2.2.1 Small capacity gas turbines - Those gas turbines under 5,000 HP are classified as small capacity gas turbines. In this size category, the majority of units are used for and divided fairly evenly between gas compression and stand-by electrical generation. Over 70 percent of all units sold in the United States in 1974 were under 5,000 HP. However, these units represent less than 5 percent of the total HP capacity of all units sold in the United States in that year.

Under 1,000 HP, the AiResearch Manufacturing Company, a division of Garrett Corporation, dominates the market. In the last two years, the company has increased its domestic sales by over 300 percent by orienting its sales to OEM[1] contracts.

From 1,000 to 5,000 HP, Solar Division of International Harvester has increasingly dominated the market for at least the past five years. The overwhelming majority of its units are either gas compression or stand-by electrical generation sets. Solar's main competition comes from Detroit Diesel Allison (a division of General Motors). An industry magazine reported that between 1970 and 1974, GM's gas turbine sales jumped 45 percent. Solar's sales during that period increased at approximately the same rate.

Avco Lycoming is the only other manufacturer actively competing for the small capacity gas turbine market. Its 1974 production was approximately 20 percent of Solar's with sales concentrated almost completely in units under 3,000 HP. Solar, Detroit Diesel Allison, and Lycoming did not sell any units over 5,000 HP in 1974. However, Solar is building a 10,000 HP turbine which should be on the market soon.

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[1] An Original Equipment Manufacturer (OEM) is one who buys an engine from an engine manufacturer and incorporates it into a product of which the engine is only a component (e.g., a portable compressor which is usually assembled by an OEM using his own housing and compressor but a purchased engine).

Of the packagers, only Ingersoll-Rand sold any units domestically in this size range during 1974. The company sold twenty-two 4,000 HP Rolls-Royce gas turbines in 1974, thirteen of them domestically. Cooper-Bessemer advertises a unit in this size range (also Rolls-Royce) but there is no record of the company making any sales under 12,500 HP since 1972. The recent drop in domestic gas compression sales probably accounts for the limited Cooper-Bessemer sales.

3.1.2.2.2 Medium capacity gas turbines - Mid-range gas turbines, between 5,000 and 20,000 HP, are employed in a variety of applications with the majority utilized in stand-by generation and pipeline compression and pumping. However, this is the size range where gas turbines are diversified in their applications. These include mechanical drive, shaft power, and a variety of other industrial uses. This category accounted for only a small part of the total domestic sales in 1974. As electric generating units have increased in size over the last five years, the relative demand for this size gas turbine has dropped.

General Electric, TP&M, and Westinghouse are the only U. S. manufacturers of gas turbines that were active in this range in the last five years. In 1974, only GE and TP&M sold any units domestically. Three packagers, Ingersoll-Rand, Dresser-Clark, and Cooper-Bessemer exported 21 units in 1974. This is more units than the major manufacturers combined sold in 1974 and these packagers have been actively competing over the past five years.

This size range is one area where foreign manufacturers have made inroads into the American market since 1970. Private industry purchased five units totalling 69,300 HP in 1974 for electric power generation, and in the last five years, several foreign firms have sold American oil companies gas turbines to be used in pipeline compression.

3.1.2.2.3 Large capacity gas turbines - Gas turbines over 20,000 HP or approximately 15 megawatts (MW) are almost exclusively used in electric generator drive applications. The largest single unit gas turbines in this category reach sizes that approach 100 MW or approximately 134,000 HP. Between 15 MW and 100 MW, active market competition occurs between five American firms. These firms are General Electric, Turbo Power and Marine, and Westinghouse; in addition, Curtiss-Wright Corporation and Turbodyne Corporation package and sell turbines under their own names.

Because of the considerable expense and complexity of the equipment involved, these five companies produce gas turbines only on a custom order basis. The lead time for delivery in 1974 was from 6 to 9 months, but as of May 1975, it had climbed to 24 months. This is much longer than that required for the smaller, mass-produced models discussed earlier.

The custom building of this large equipment allows for variations in rated generating capacity for each model sold by the five firms. The active competition in this market has produced increased efforts to improve fuel efficiency levels. In addition, newer techniques have been developed, such as combined cycle technology.

3.1.2.2.4 Multiple units and combined cycle - Since the late 1960's, growth in total HP installed has been accompanied by a substantial drop in the number of units produced. This trend is partly due to the industry's increasing production of multiple tandem and combined cycle installations. These installations have the capacity to produce well over 100 MW with one application having a total rating of 1230 MW.[3] Table 3-1 indicated that five companies market individual equipment configurations that have a generating capacity of over 350 MW. Multiple unit installations with these large capacities are increasingly being installed for mid-range (3,000 to 5,000 hours) and baseload capabilities though they still represent only a small portion of the total market.

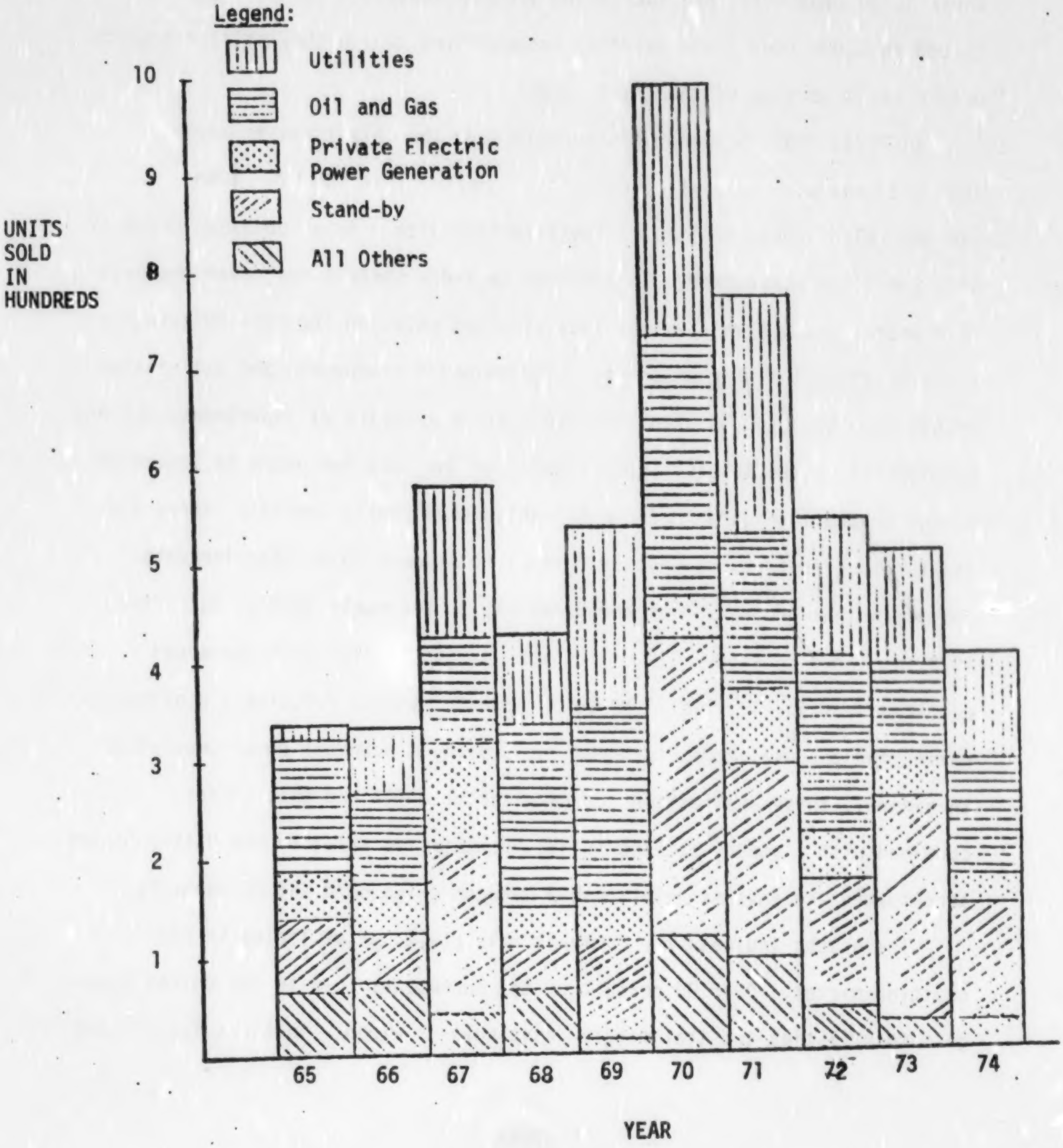
In 1974, TP&M, GE, and Westinghouse were the only firms to market installations with generating capacities greater than 100 MW. TP&M was the solid leader selling 17 installations with a total combined capacity of 3.9 million horsepower. In addition to these three firms, over the past five years, Pratt & Whitney has also recorded sales in the 100+ MW category.

3.1.2.3 Historical Market Trends - The domestic production and utilization of gas turbines blossomed in the 1960's after a period of experimentation and development in the 1950's. This growth for the last ten years is indicated by the graphs in Figures 3-2 and 3-3 which show yearly domestic orders for gas turbines in terms of units and total horsepower sold. Earlier data indicates that production expanded sharply in the early 1960's, but after 1970, as shown in Figure 3-2, the number of gas turbine units produced declined drastically. Total HP sold, however, stayed relatively constant, reflecting the use of gas turbines for utility electrical power generation. In general, these applications require larger units and this factor, combined with a steady decline in gas transmission sales in the United States, accentuated the inverse relationship between units sold and HP capacity.

The graphs indicate the dominant role played by utilities in the existing market. Over 90 percent of the horsepower sold in the United States goes to utilities and that margin continues to widen. Turbines for oil and

Figure 3-2.

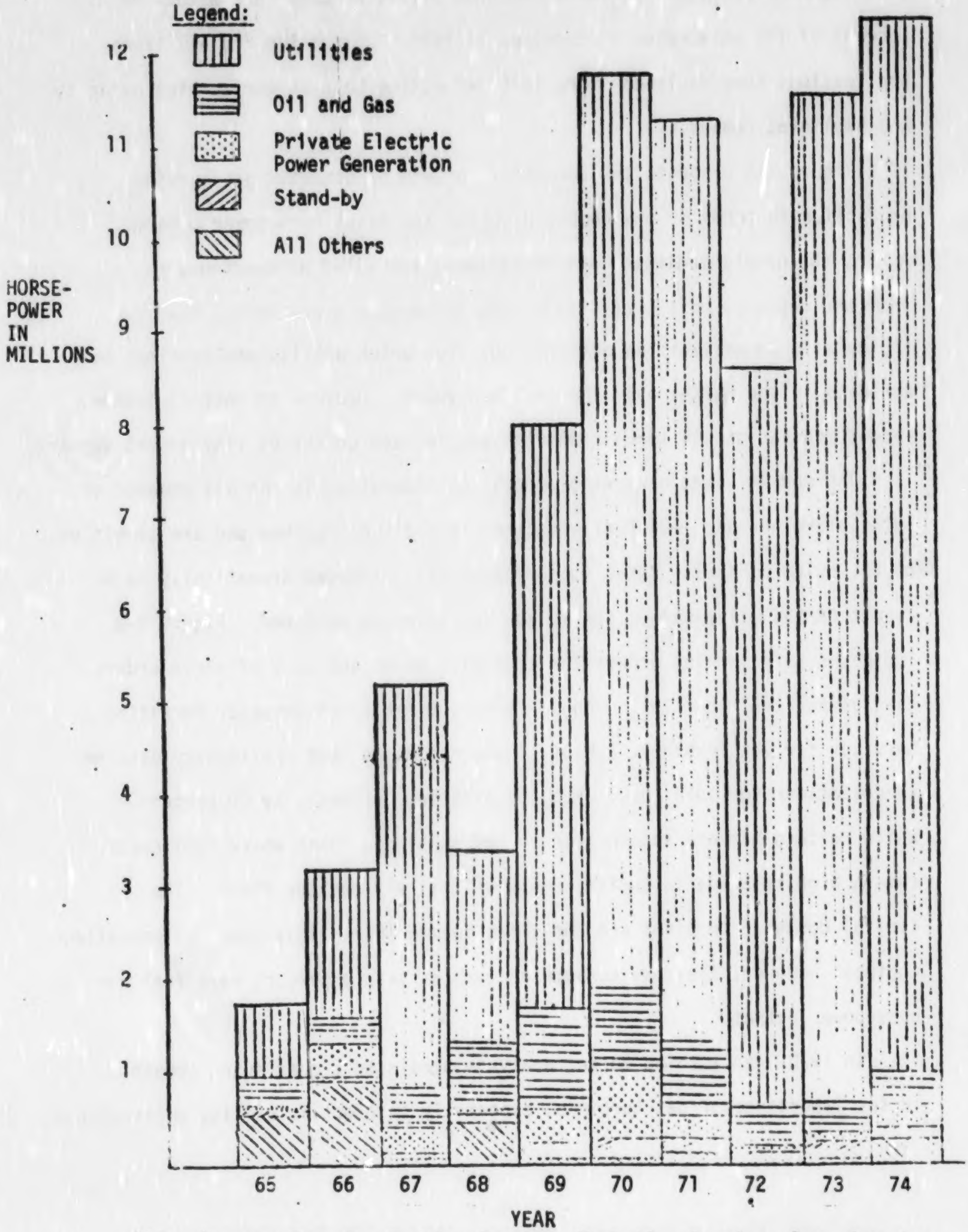
YEARLY DOMESTIC ORDERS FOR STATIONARY GAS TURBINES BY APPLICATION, GIVEN IN UNITS, BY YEAR



SOURCE: Adapted from Gas Turbine Computer Marketing Service, Inc., data base.

Figure 3-3

YEARLY DOMESTIC ORDERS FOR STATIONARY GAS TURBINES, BY APPLICATION, GIVEN IN HORSEPOWER, BY YEAR



CE: Adapted from Gas Turbine Computer Marketing Service, Inc, data ba gas, stand-by electricity generation, and utilities comprised almost 90 percent of the unit sales of turbines in 1974. Unit sales for all three applications have declined since 1970 reflecting both larger turbine capacity and declining demand.

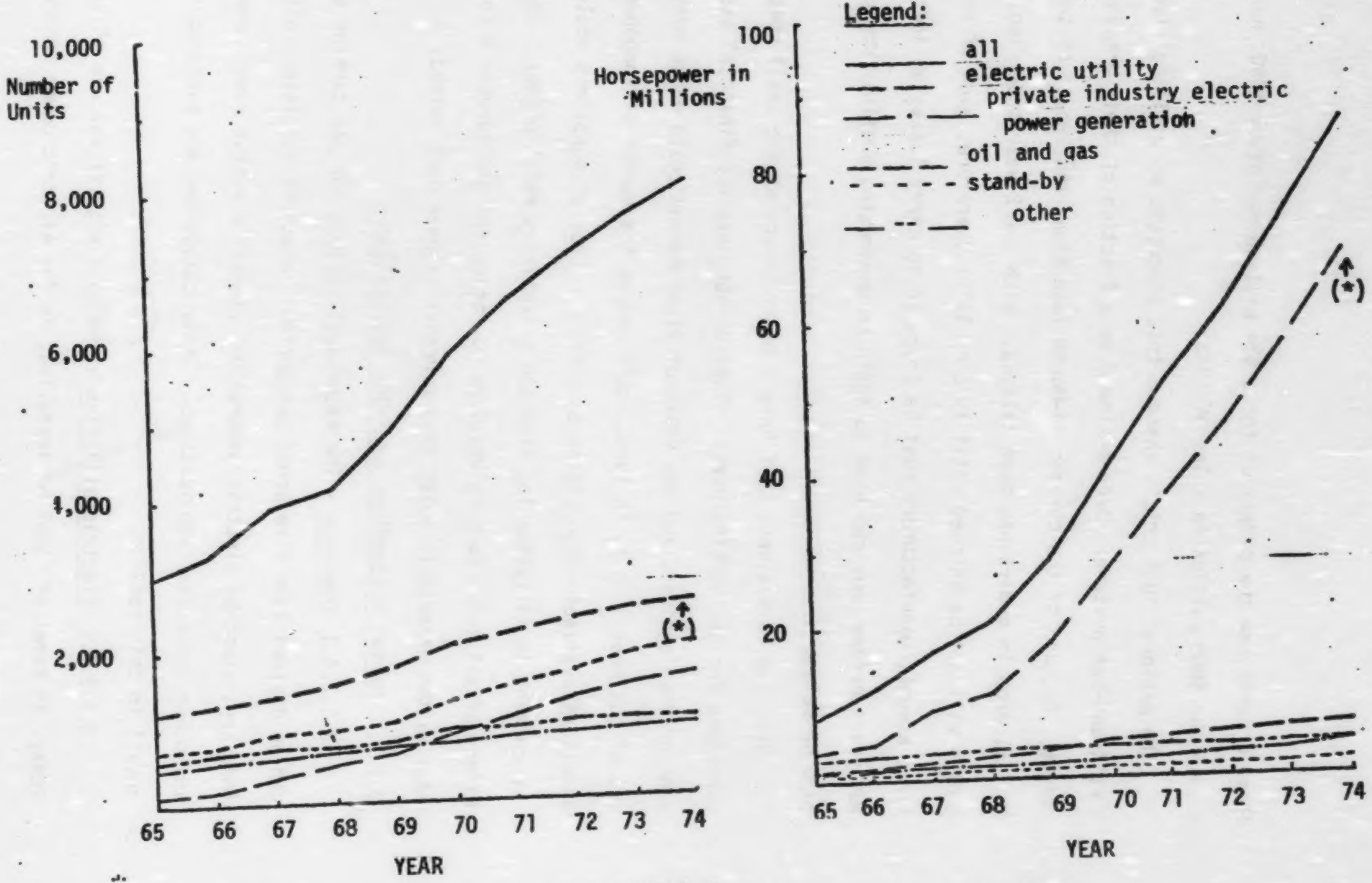
Figure 3-4 presents the cumulative growth of domestic gas turbine production in terms of the number of units and total horsepower ordered. Again, the graphs indicate that while total installed HP continues to increase rapidly, unit orders have been decreasing since 1970. The figure also emphasizes the dominant position which utility applications have achieved in the market over the past ten years. Because of this dominance, recent innovations in turbine design have focused on the utility market segment.

One market event deserving special consideration is the oil embargo of 1973. Turbines are less fuel efficient than diesel engines and are sensitive to the price of fuels. When the price of oil increased dramatically as a result of the oil embargo, the market for turbines declined. Figure 3-3 indicates 1974 turbine orders for 12.4 million HP but many of these orders have since been cancelled. Robert Farmer, Director of Computer Marketing Services for Gas Turbines, C.M.S., Inc., estimates that preliminary data on cancellations for 1974 could drop the existing estimates by 50 percent. While certain markets remain open to gas turbines, those where fuel costs remain high relative to capital costs will likely decline first. The capital costs of turbines are one-third to two-thirds less than reciprocating engines; however, they may consume as much as thirty percent more fuel per horsepower output.

In 1972, gas turbine market analysts predicted a continuing growth in the importance of gas turbines, especially in electric utility applications.

Figure 3-4.

CUMULATIVE DOMESTIC ORDERS FOR STATIONARY GAS TURBINES BY APPLICATION, GIVEN IN UNITS AND HORSEPOWER



(*) 1974 orders may decline due to cancellations.
SOURCE: Adopted from Gas Turbine Computer Marketing Services, Inc., data base.

Those experts saw the potential for a $2+ billion market by 1980, nearly triple the $800 million in sales of 1972.

The national fuel crisis changed this prognosis as utilities took a wait-and-see posture. Orders slowed to a fraction of their previous levels. Because of the current 24-month lead time, the impact of the initial drop in orders has been delayed. With the 2-year order lag, 1976 orders will not be shipped until 1978 or 1979; therefore, even if the market turns around, manufacturers must be prepared to carry losses an additional two years before they can hope to again be generating profits from gas turbine sales.

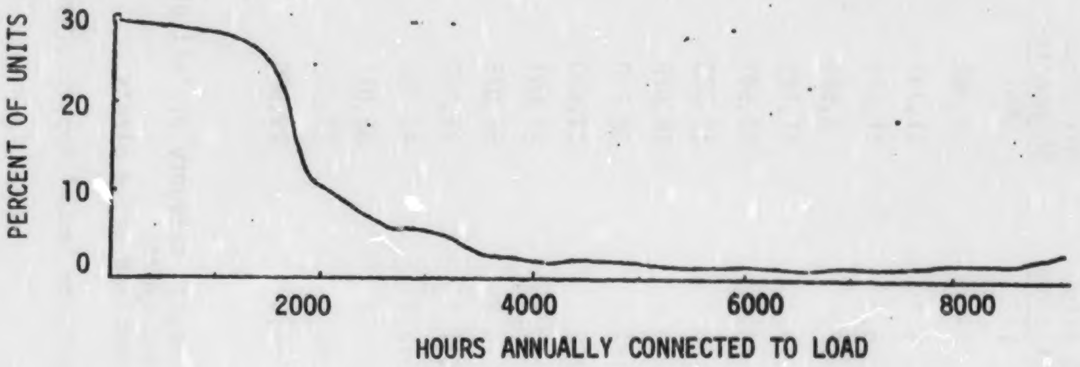
This lag in business will have a major impact on the small component suppliers for gas turbine parts. They do not have the financial base of a GE or Westinghouse, and two years of slack demand could cause many to go out of business. This, in turn, would reduce the number of component suppliers available when orders do pick up. Lack of suppliers would lengthen the current installation lag time and potentially raise prices. The net effect of all these factors would be to reduce the gas turbine's competitive edge in the markets to which they currently appear best suited.

3.1.2.4 Market Projections by Major Application -

3.1.2.4.1 Overview - The major applications for gas turbine equipment can be divided into five broad categories: electric utilities; oil and gas industry; stand-by electric generation; private electric power generation; and other industrial applications. These categories are analyzed in greater detail in this section.

3.1.2.4.2 Electric utilities - The most significant use of gas turbines today, in terms of total HP installed, is for electric power generation by electric utilities. The relative position of utility applications to other gas turbine applications has increased more than 10-fold over the past 10 years, an indication that gas turbines have become an essential part of the nation's power base.

Gas turbines are considered most popular as peak-load equipment. A random sample of gas turbine applications taken from a 1972 Federal Power Commission source indicates the following relationship between hours run connected to load and percentage of units installed. Most units run less than 2,000 hours per year.



Shipments of gas turbines to domestic electric utilities are presented in Table 3-5. Shipments peaked in 1971 after a dramatic 13-year climb. The major decline in shipments, however, began in 1974 after 1973 fuel price increases discouraged turbine buyers.

Estimated 1975 shipments are 62 percent below 1974 levels and 72 percent below 1973 levels. In 1973, 20 percent of all orders for new generating machines were for gas turbines, but by 1975, this figure had fallen to only seven percent.

The cost of turbines per kolowatt correlates with volume because the fixed costs of turbine manufacturing (e.g., set-up costs) are so high. The

Table 3.5

U.S. GAS TURBINE SHIPMENTS TO DOMESTIC ELECTRIC UTILITIES

	Year	Gas Turbine Shipments (MW)	Average $ per KW ($)	$ Value Millions ($)	All other Shipments[1] (MW)	Total of All Shipments (MW)	Gas Turbine Percentage (%)
	1959	0	--	--	12,043	12,043	--
	1960	45	96	4.3	11,791	11,836	0.4
	1961	100	88	8.8	11,812	11,912	0.8
	1962	210	84	17.6	8,669	8,879	2.4
	1963	220	87	19.1	11,189	11,409	1.9
	1964	326	71	23.1	12,597	12,923	2.5
	1965	419	72	30.2	12,323	12,742	3.3
	1966	598	73	43.6	14,628	15,226	3.9
	1967	2,291	71	162.7	30,770	33,061	9.9
	1968	3,522	66	232.5	24,630	28,152	12.5
	1969	3,725	67	249.6	25,891	29,616	12.6
	1970	5,780	72	416.2	26,028	31,808	18.2
	1971	7,815	68	531.4	35,027	42,842	18.2
	1972	7,217[2]	71	512.4	43,308	50,525	14.3
	1973	7,417[2]	65	482.1	30,011	37,428	19.8
	1974	5,500[2]	68	374.0	27,600	33,100	16.6
est.	1975	2,100[2]	75	165.0	29,500	31,600	6.6

[1] Gas turbine MW's shipped based on peak ratings: category of "all other" includes fossil steam and steam portion of combined cycle, nuclear and hydro.

[2] Includes only the gas turbine portion of combined cycle plants.

Source: Data Sources, Edison Electric Institute Annual Reports, Gas Turbine World Studies, April, 1975.

lowest cost per kw was recorded in the early 1970's as sales reached their peak. However, the estimated figures for 1975 show an average cost per kw to be around $75, the highest in 12 years.

A survey of domestic utilities presented in Sawyer's 1975 Gas Turbine Catalog gives some indication of future sales in this sector. Their figures showed no appreciable increase in demand for gas turbines as a percentage of total generation mix between 1975 and 1985. More than half of all utilities anticipate that gas turbines will represent no more than 10 percent of the future generation mix. The primary reason given is the increased cost of fuel and the unpredictability of oil and gas supplies.

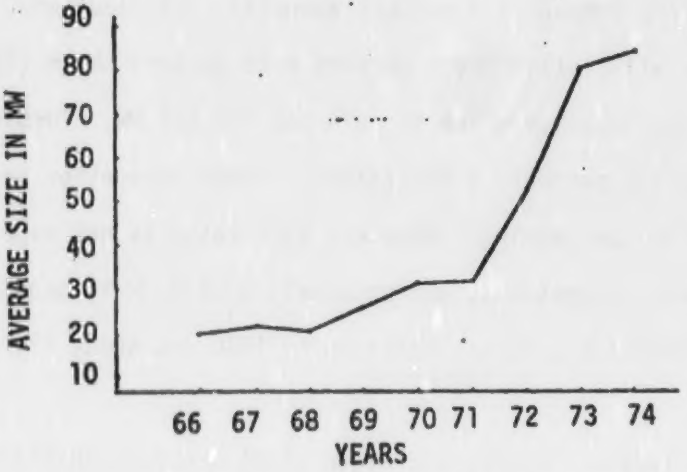
The analysis assumes that total generating capacity will grow at the same rate as electricity demand (6.4 percent annually). In addition, it assumes that turbines will retain their current 8.25 percent share (39,300 MW) of the total generating capacity which in 1974 was 475,897 MW. Turbines currently supply only 1.6 percent of the nation's power generation because they are used primarily for peaking. However, this ratio is not expected to change. Based on these assumptions, approximately 57,000 MW in generating capacity would be attributable to gas turbines by 1980 and about 77,700 MW by 1985.

A 1974 study by Temple, Barker, and Sloan (TBS) provides an alternative projection. They predict that by 1985 "peakers" will make up only 7.9 percent of the electric generation mix. The TBS analysis concurs with EEA projections for net generating output from gas turbines in 1985. However, they also predict that gas turbines will decrease their share of the generation mix over the next two years and by 1985 will account for slightly less than

64,000 MW of generating capacity. The increase in generating capacity would, therefore, be only 24,700 MW or 40 percent less than that indicated by EEA projections.

The TBS study therefore implies longer yearly operating time for gas turbine equipment. However, this assumption contradicts the 1975 user survey information on generation mix done by Sawyer's Catalog.

To estimate unit sales from MW output data, it is necessary to determine the size distribution of the units being sold. Energy and Environmental Analysis, Inc., (EEA) data indicates the following trend in average gas turbine unit sizes for utility applications over the past ten years:



"Units" in this case include combined cycle and tandem gas turbine configurations. Assuming that the average unit in the next ten years will be at least 80 MW, the following table indicates the range of market projections:

	1980		1985	
	Units	MW	Units	MW
EEA Assumption	221	17,700	481	38,400
TBS Assumption	140	11,400	309	24,700

3.1.2.4.3 Oil and gas industry - Gas turbines in the oil and gas industry are used almost exclusively for oil and gas transmission and extraction. The gas turbines fit the needs of pipeline applications because they are available in a variety of sizes and are suitable for semi-remote control operation. Most of these turbines run continuously, 8000 hours per year.

Solar has dominated the oil and gas transmission business in recent years, especially in the smaller size ranges under 5,000 HP. In the 5,000 - 20,000 HP range, or the upper end of gas and oil transmission applications, foreign firms historically have been successful. In 1970, ten percent of the gas turbines purchased by the oil and gas industry in the United States were manufactured by foreign firms. In recent years, however, foreign sales have fallen off to virtually nothing.

Gas turbines have a considerable capital cost advantage over reciprocating engines. FPC data shown below indicates a reciprocating engine costs 33 63 percent more than a comparable gas turbine.

	$/Horsepower		
Application	Turbine	Reciprocating	Ratio of Reciprocating to Turbines
New transmission compressor stations	252	337	1.33
Additions to transmission compressor stations	274	367	1.34
New field and storage compressor stations	353	565	1.60
Additions to field and storage compressor stations	288	470	1.63

Diesels are used because they require less fuel than turbines to produce the same HP and consequently have lower operating costs. Another advantage of gas turbines over internal combustion engines is that they can be operated by remote control and need not be inspected very often. Addition of NO_x controls using water injection could eliminate this advantage since the water treatment equipment would have to be inspected periodically. An informed industry source indicates, however, that most users of pipeline transmission equipment see fit to have on-site inspections at least once a week. This factor would seem to discount the idea that installation of NO_x controls are detrimental to the future competitive position of turbines.

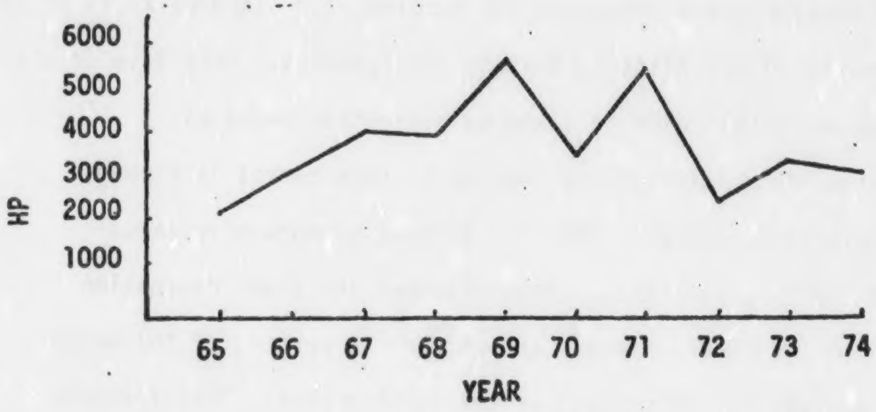
Calculation of future domestic gas turbine demand to 1985 in the oil and gas extraction and transmission business will be based upon estimated oil and gas production scenarios. The Project Independence Blueprint (PIB) provides two scenarios: a $7 per barrel business-as-usual (BAU) format and an $11 per barrel accelerated development (A-D) format. The PIB average annual compound growth rate estimates for the 1974 to 1985 period are as follows:

	Oil	Gas	Average
BAU	1.14%	4.30%	2.72%
A-D	6.03%	7.35%	6.7%

A range for potential future production can therefore be assumed to be between 1.14 percent and 7.35 percent.

EEA data indicates that total gas turbine horsepower installed through 1974 for the oil and gas industry, less electric power generation, amounted to approximately 6,831,000 HP. Using the average growth rate under the business-as-usual format, the projected additional demand for the 7-year period from 1974 to 1980 inclusive would be 1,411,000 HP and for the 12-year period through 1985 would be 2,592,000 HP. Using the accelerated development format, the projected additional 7-year demand would be 3,924,000 HP and the 12-year demand would be 8,073,000 HP. The growth rate in 1974 was 5.2 percent which would produce an estimate between the two already calculated.

The following graph indicates the fluctuations in the average size of gas turbines ordered for use by the oil and gas industry for extraction or transmission purposes:



Though the trend appears to be toward smaller individual units, it is reasonable to assume that the average size unit over the next 10 years will be at least 3000 HP. Using the 3000 HP figure, it is possible to calculate the number of units to be sold. Under the business-as-usual format, approximately 470 units would be sold between 1974 and 1980, and 864 between 1974 and 1985. Under the accelerated development format, there appears to be a market for almost 1300 new units through 1980, and 2700 new units through 1985.

3.1.2.4.4 Stand-by electric generation - Small electric generator sets make up a considerable number of all gas turbine sales under 5,000 HP. In the United States, the majority of these sets are used for stand-by electric generation, with the telephone companies constituting the principal users. In addition, some gas turbine stand-by units are used by hospitals and small municipalities. These turbines are operated an average of between 75 and 200 hours per year.

In a stand-by application, the annual operating time of the generating equipment is typically under 100 hours. The user requirements are quick starts, low capital costs, ability to burn a variety of fuels, high reliability and easy maintenance. The gas turbine meets these requirements.

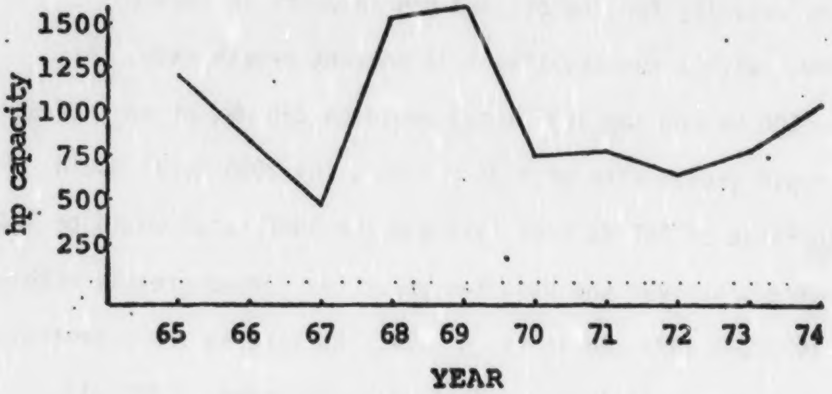
The American manufacturers producing gas turbines for stand-by applications include Solar, Detroit Diesel Allison, Garrett and Lycoming. Only five or six different turbines are widely used in stand-by generation packages.

The competition from reciprocating engines in this market is strong, especially for applications under 1,000 HP. A study by market research specialists Frost and Sullivan, Inc., estimates that for power generation units under 3,000 kw (4,020 HP) diesels account for 68 percent of the market and by 1985 will account for 76 percent of the total market. They indicate that the market in general will grow at a 12.6 percent annual rate with diesels growing at a 15.3 percent annual rate. Gasoline units in this market will only grow at a 6.2 percent annual rate.

The Frost and Sullivan study noted that trying to estimate the growth rate of gas turbines in small generator sets was "one of the great enigmas of our investigation". Nevertheless, they predicted a conservative annual growth rate for turbines of nearly 10 percent.

The 1974 EEA data indicates that the total horsepower capacity for stand-by units is 1,256,842 HP (or 937 MW). Using a 10 percent annual compounded growth rate, there would be an increase from 1974 to 1980 inclusive of 1,193,000 HP (or 888 MW) and from 1974 to 1985inclusive of 2,695,000 HP (or 2000 MW).

The average size stand-by electric generator sets has remained relatively small over the past ten years as the following grapth indicates:



Assuming the average size unit will be approximately 1,000 HP over the next ten years, approximately 2,700 units will be marketed for this application, with 1,200 of them sold by 1980.

3.1.2.4.5 Private industry electric power generation - Electric power generation by private industry was the third most important use of stationary gas turbine equipment in 1974 in terms of HP. Large industrial complexes and refining facilities consume considerable amounts of electricity. Turbines will run between 4,000 and 8,000 hours per year, depending on the application. This category has been divided into electric generation by the oil and gas industry and electric generation by "other" private industries.

A significant increase in generating capacity occurred in 1974 in applications for the oil and gas industry. The advantages of gas turbines over steam electric plants in terms of reliability and flexibility appear to give gas turbines a very strong market position in this industry.

A projection of future demand for gas turbine electric generating equipment in the oil and gas industry can again be based on the Project Independence Blueprint high and low annual growth rates presented earlier. Under this scenario, the range would be between 1.14 percent and 7.35 percent. The total generating capacity for the oil and gas industry in 1974 was approximately 460 MW. With a conservative 1.14 percent growth rate, the 1980 total would be 500 MW and the 1985 total would be 536 MW, or an increase of 76 MW. With a rapid growth rate of 7.35 percent, the 1985 total would be 1003 MW or an increase of 543 MW from 1974 and the 1980 total would be 704MW.

The average size unit over the past ten years has varied greatly with units averaging 1 MW in 1972 and 1973 and 10 MW in 1974. Estimating the potential for gas turbine installations in this application is therefore difficult. However, using a 5 MW average size unit, the total installations through 1980 would be 8 units under the conservative growth rate and 49 with the accelerated development format, and through 1985 the number of units would range from 15 to 108.

For electrical generation from other industrial sources, an estimation of the annual growth rate of gas turbine applications can be based upon the estimated annual growth rate of the industrial sector. Using the summer 1975 forecast of the United States economy by Data Resources, Inc., a 5.979 percent annual compounded growth rate can be established from the industrial production index. EEA data indicates the total industrial electrical generating capacity in 1974 was approximately 1,900 MW. Using a 6 percent annual growth rate, the 1985 generating capacity should reach 3,600 MW or an increase of 1,700 MW.

The average size installation over the past ten years has followed the same highly variable pattern as did the oil and gas industry, though with a few units each year approaching the 20 MW level. However, for simplicity, the 5 MW unit will again be used as the average size unit for future installations indicating approximately 340 new gas turbines will be installed in the industrial sector for electric power generation.

3.1.2.4.6 Other industrial applications - Industrial applications for gas turbines include various types of mechanical drive and air compression equipment. These applications reached their peak in the late 1960's and have fallen to relative insignificance in recent years, both in terms of units and horsepower installed.

Gas turbines should meet continuing stiff competition from diesel engines for most of the smaller industrial applications to which they are now applied. However, gas turbines do have several attractive attributes for industrial drive applications. These are their small size, high horsepower-to-weight ratio, ability to burn a variety of fuels, reasonable reliability, availability in self-contained packages, power without warm-up quality, lack of cooling water requirements, vibration-free operation, and low installation costs. This flexibility is especially valuable where operating requirements may run from 2,000 to 8,000 hours per year.

Again, the estimated demand for industrial gas turbine applications, less electrical power generation, can be based upon the projected annual growth of the industrial sector. EEA data indicates a total of 1,146,589 HP used through 1974 for industrial drive type gas turbine applications. With the 6 percent annual growth rate, 2,175,000 HP of drive will come from gas turbines by 1985 or an increase since 1974 of approximately 1,029,000 HP. However, this is probably a high estimate considering the few installations in recent years.

Estimating the average size unit and the total new units through 1985 is of dubious value due to the great variety in size of installations. There were 13 installations averaging 1,200 HP in 1972 and 3 installations averaging 19,500 HP in 1974.

3.1.2.4.7 Summary of market projections - Aggregation of the results from the five market sectors analyzed provide a profile of the turbine market beyond 1974. Tables 3-6 and 3-7 summarize the market projections through 1980 and through 1985, respectively.

Utilities will continue to dominate the market with at least 80 percent of the HP requirements. Stand-by and oil and gas industry applications will account for over 80 percent of unit sales.

If the market projections are annualized, then the decline from 1974 order levels becomes more obvious. Domestic orders for turbines in 1974 totalled 12.2 million horsepower and 434 units. The 1974 through 1985 projection produces an annual order rate of 3.8 - 6.0 million HP and 383 - 574 units. This represents a stabilization in unit sales which have been declining since 1970 but a drastic decline in HP sales. Preliminary information about order cancellations in 1974 seems to indicate that the downward adjustment in HP demand has already occurred although it is not reflected by existing data.

The market projections also reflect tremendous uncertainty. Capacity and unit sales could vary by 75 percent and 90 percent, respectively, depending on growth in power consumption and the future price of oil. However, even under the most optimistic assumptions, growth rates of the late 1960's

Table 3-6

1974 - 1980 MARKET PROJECTIONS

Application	Increase in Capacity (M HP)		Increase in Units	
	High	Low	High	Low
Utilities	23.70	15.30	221	141
Oil and Gas	3.92	1.41	1300	470
Stand-by	1.19	1.19	1196	1190
Private Electric Power Generation				
-- oil and gas	.33	.05	49	8
-- other industry	1.06	1.06	159	159
Other Industrial Applications	.48	.48	N/A	N/A
TOTAL	30.68	17.49	2919	1967

Table 3-7

1974 - 1985 MARKET PROJECTIONS

Application	Increase in Capacity (M HP)		Increase in Units	
	High	Low	High	Low
Utilities	15.50	33.10	481	309
Oil and Gas	8.04	2.59	2700	860
Stand-by	2.69	1.69	2690	2690
Private Electric Power Generation				
-- oil and gas	.73	.10	108	15
-- other industry	2.28	2.28	340	340
Other Industrial Applications	1.02	1.02	N/A	N/A
TOTAL	66.26	41.78	6319	4214

and early 1970's are not expected to reappear unless new turbine developments occur or new applications are discovered. Table 3-8 presents a summary of the methodology used in calculating the growth projections.

3.1.2.5 Balance of Trade Position - Over the last five years, gas turbines have improved their already positive balance of trade position. Exports have increased considerably and have helped to make up for the reduction in domestic sales caused by increased fuel prices. Imports, on a direct sales basis, have fallen to relative unimportance.

The export market for U. S. manufactured gas turbines was about 5.46 million HP in 1974, or about 30 percent of total U. S. gas turbine output. Exports for the last five years were as follows, as indicated by EEA data:

	1970	1971	1972	1973	1974
Units	194	257	198	291	372
HP (000)	914.4	1572.5	1837.4	7079.6	5458.4

Although growth has not been steady, the total HP sold has more than quadrupled from 1970 to 1974, while the number of units sold has increased by about 90 percent. The difference in the magnitude of the HP and unit increases reflects the trend toward larger individual units used in electric power generation.

The U. S. export market for gas turbines is dominated by five firms: General Electric, Avco Lycoming, Solar, Turbo Power and Marine, and Westinghouse. These five firms accounted for about 87 percent of the units exported in 1970 and about 90 percent of the units exported in 1974. Their market control appears independent of yearly export fluctuations.

The three firms shown below are the largest exporters in terms of HP and unit sales:

Table 3-8A

METHODOLOGY FOR MARKET PROJECTION

The above calculations were done using the following methodology:

(1) Capacity in place was determined for 1972 or 1974;

(2) A range of annual growth rates was determined for the each of the sectors using turbines;

(3) The increase in capacity was calculated using the formula:

$$\Delta X = X_o (1+r)^n - X_o$$

where ΔX = increased turbine capacity between base year and projection year
X_o = capacity in place in the base year
r = annual demand growth rate
n = number of years between base year and the projection year

(4) An average unit size (U_o) was determined based on historical data on turbine sizes;

(5) The number of new turbines was calculated using the formula:

$$\text{number of new units} = \frac{\Delta X}{U_o}$$

(6) Because a turbine will last a minimum of 20 years and because over 90 percent of turbine capacity was installed after 1965 (see Figure 3-3), the replacement market was assumed to be a negligible portion of new turbine sales.

	% of Total HP	% of Total Units	Average Unit Size (HP)
General Elec.	40	23	27,276
Solar	8.5	45	2,800
Westinghouse	35	5	94,260

GE units are used primarily for electric utility power generation. Solar supplies the smaller gas compression, generator drive, and oil and water pump markets. Westinghouse units are used primarily for generator drive. Of the other two firms, Avco Lycoming Division has exported smaller units used mostly for marine purposes whereas Turbo Power and Marine has exported larger units used mostly for peak load requirements. Firms dealing in larger units seem to have experienced a relatively stable growth in their exports while the exports of those dealing in smaller units have fluctuated from year to year along with the total export market.

The Department of Commerce has placed the dollar value ($ million) of turbines shipped as follows:

	1970	1971	1972	1973	1974
Gas Turbine Generator Sets Assembled or Unassembled	$13.6	$30.6	$21.9	$98.2	$150.6
Gas Turbines and Parts for Mechanical Drive, NEC[1]	$51.0	$71.1	$86.9	$137.0	$153.5
TOTAL	$64.6	$101.7	$108.8	$263.2	$304.1

According to these figures, the dollar value of gas turbines, gas turbine generator sets, and parts for mechanical drive shipped abroad has more than quadrupled since 1970. Turbine shipments now contribute over $300 million to the balance of payments and are growing rapidly.

---

[1] NEC = Not elsewhere classified

During this same period, imports of gas turbines into the United States have been steadily declining. The total HP imported was only about 3 percent of the 1970 total. The data compiled by EEA, which does not include foreign manufactured equipment sold to American packagers, is as follows:

	1970	1971	1972	1973	1974
Units	40	40	18	8	3
HP (00)	385.8	408.7	275.9	13.1	29.5

Doubtless, the devaluation of the U. S. dollar influenced the increasing U. S. exports and reduced imports of gas turbines. The predominance of U. S. exports seems likely to continue. Department of Commerce figures concerning the shipments of gas turbines and parts for mechanical drive (not elsewhere classified) which were cited above show that $110,847,457 worth of merchandise in this grouping have already been shipped in the first half of 1975. If this rate of shipments continues, the 1975 dollar value exported will surpass 1974.

3.2 PROCESSES AND THEIR EMISSIONS

3.2.1 Processes

3.2.1.1 The Simple Cycle Gas Turbine - A gas turbine is a rotary engine which consists typically of one or more compressor stages, one or more combustion chambers where liquid or gaseous fuel is burned, and one or more turbines to drive the compressor(s). For stationary applications, a power turbine is included to drive the load. There are various configurations of these basic gas turbine components. Figure 3-5 presents a block diagram showing a typical simple cycle gas turbine. Figure 3-6 presents a cut-away view of a dual spool gas turbine with can-annular combustors and a free power turbine. It is called a dual spool turbine because the low pressure compressor and the high pressure compressor stages consist of two mechanically independent rotor systems and the

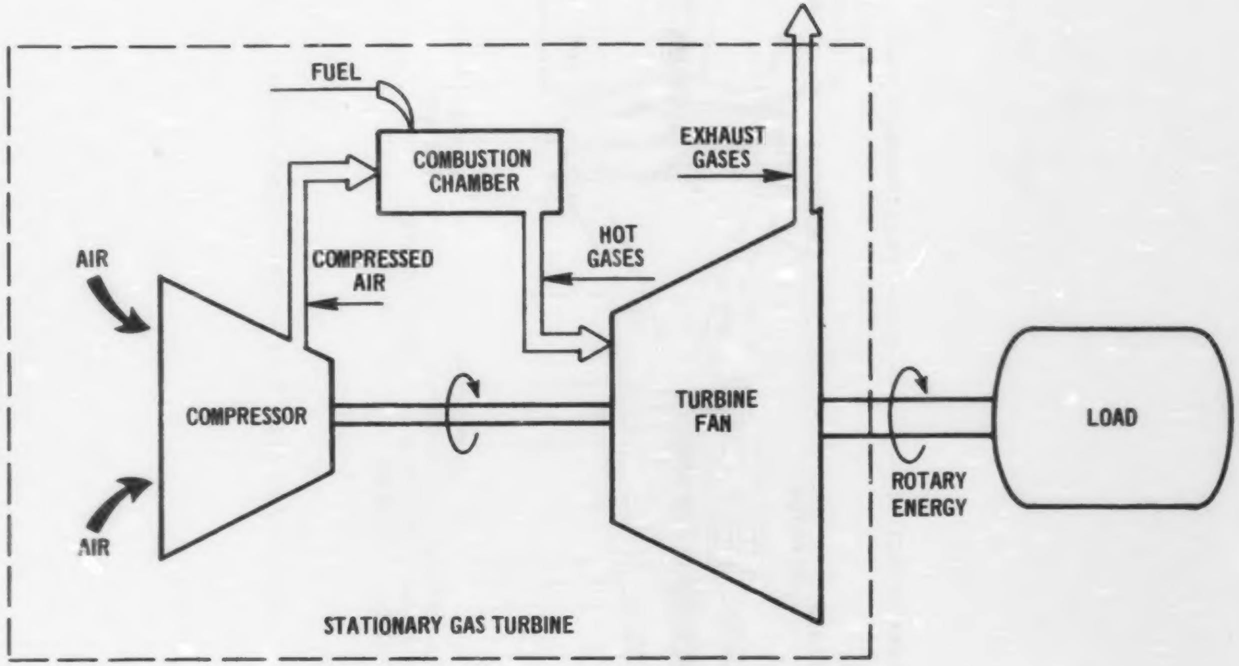
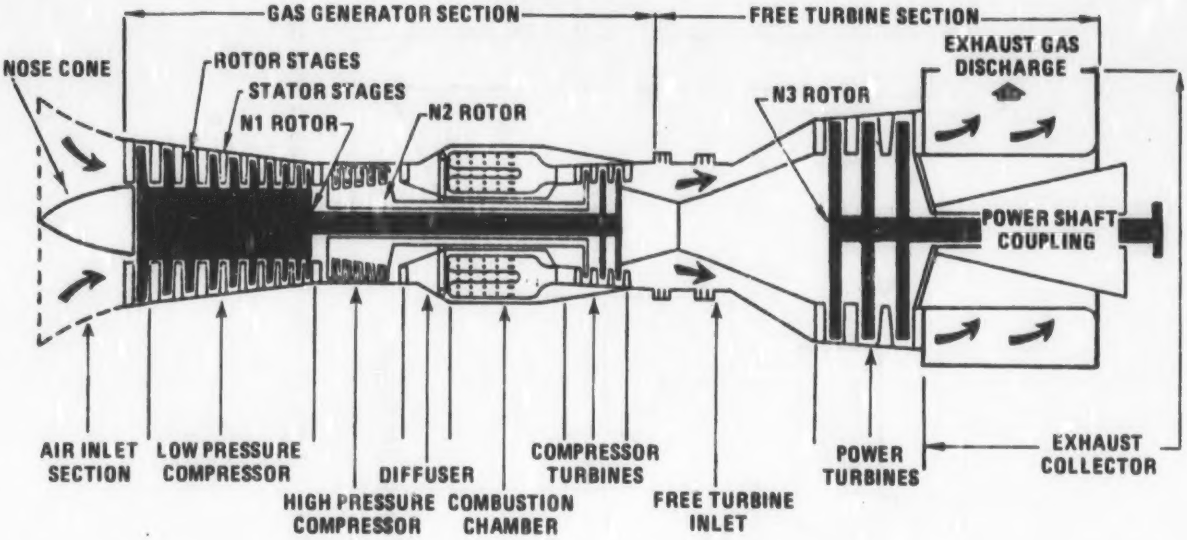

FUEL
COMBUSTION CHAMBER
EXHAUST GASES
HOT GASES
AIR
COMPRESSED AIR
COMPRESSOR
TURBINE FAN
LOAD
AIR
ROTARY ENERGY
STATIONARY GAS TURBINE

Figure 3-5. Typical simple cycle gas turbine.



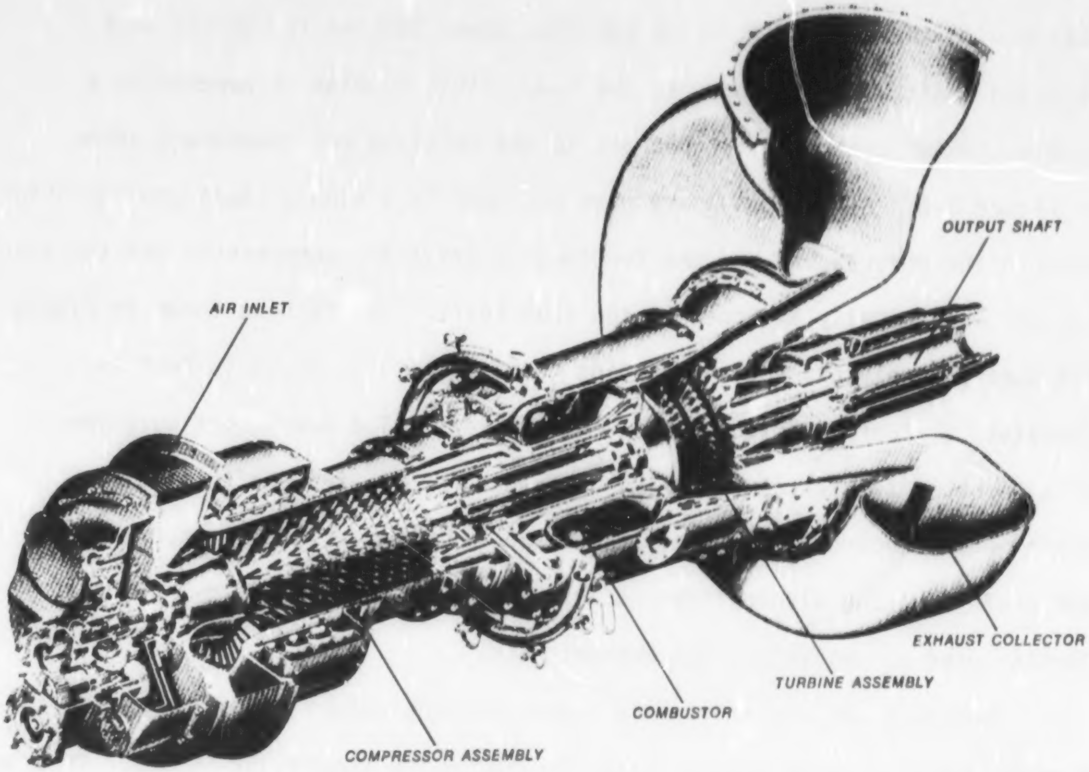
GAS GENERATOR SECTION
FREE TURBINE SECTION
NOSE CONE
ROTOR STAGES
STATOR STAGES
N1 ROTOR
N2 ROTOR
N3 ROTOR
EXHAUST GAS DISCHARGE
POWER SHAFT COUPLING
AIR INLET SECTION
LOW PRESSURE COMPRESSOR
HIGH PRESSURE COMPRESSOR
DIFFUSER
COMBUSTION CHAMBER
COMPRESSOR TURBINES
FREE TURBINE INLET
POWER TURBINES
EXHAUST COLLECTOR

Figure 3-6. Cut-away view of a typical simple cycle gas turbine.

power turbine is "free" because it is not mechanically connected to either compressor rotor. The combustion chambers "cans" are arranged in an annular configuration around the gas turbine axis, hence the term can annular. Figure 3-7 shows a two shaft turbine with an annular combustor. The compressor assembly and compressor drive turbines are mechanically attached to the same shaft while the free power turbine is mounted on a separate shaft which connects to the load. This turbine is powered by a single annular combustor in contrast to the multiple can combustors shown in Figure 3-6. Gas turbines are also produced in a single shaft configuration wherein the compressors and the turbines to drive the compressors and the load are all mechanically attached to the same shaft. The turbine shown in Figure 3-8 uses a single external combustion chamber which is about 10 feet in diameter, 21 feet in height, and weighs 10 tons. The exact configuration of a turbine (type of combustor(s) used, dual spool, two shaft, single shaft, etc.) depends to some extent on the manufacturer's past experience and preference and also on the specific market application (e.g., peaking turbine used by utilities, gas pumpers, etc.).

Gas turbines are started by using electric motors, diesel engines, compressed air or another energy source to rotate the compressor in order to provide sufficient air under pressure to support combustion in the combustors. Fuel is then introduced into the combustors and burned to produce hot gas which expands across the first set(s) of turbine blades, thus providing the driving force to continue rotating the compressor(s) mounted on the same shaft. The hot gas is further expanded across the power turbine blades which convert some of the remaining energy in the gases to output shaft power for driving

Figure 3-7 Cut-away view of a simple cycle, two shaft turbine with an annular combustor.


OUTPUT SHAFT
AIR INLET
EXHAUST COLLECTOR
TURBINE ASSEMBLY
COMBUSTOR
COMPRESSOR ASSEMBLY

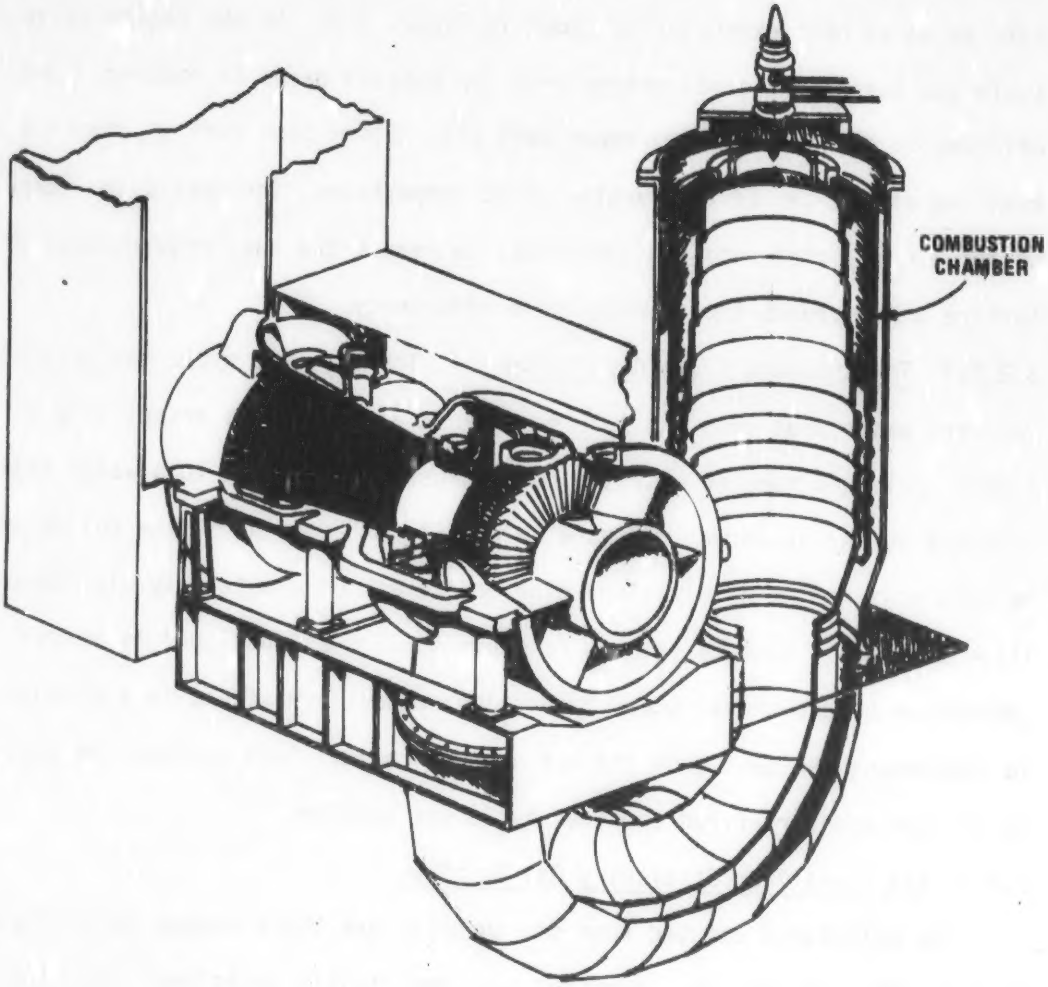

COMBUSTION
CHAMBER

Figure 3-8. Single shaft gas turbine with an external combustion chamber.

pumps, electrical generators, etc. The heat energy remaining in the gases after expansion across the power turbine can then be further utilized in steam boilers or other heat exchangers.

3.2.1.2 The Regenerative Cycle Gas Turbine - The hot combustion gases from the simple cycle gas turbine (described in Section 3.2.1) exit to atmosphere at exhaust gas temperatures ranging from about 800 to 1100°F.[6] The regenerative cycle gas turbine is essentially a simple cycle gas turbine with an added heat exchanger as shown in Figure 3-9. In the regenerative cycle gas turbine, thermal energy from the exhaust gases is recovered and utilized to preheat the compressor exit air. Since less fuel is required to heat the air to the design turbine inlet temperature, increasing the temperature of the air injected into the combustors decreases the fuel requirements of the turbine and improves the overall cycle efficiency.

3.2.1.3 The Combined Cycle Gas Turbine - The combined cycle gas turbine also recovers waste heat from the turbine exhaust gases. It is essentially a simple cycle gas turbine with the hot exhaust gases vented to a waste heat recovery boiler as shown in Figure 3-10. Steam generated in the boiler can be used in a number of ways including generation of electricity via conventional steam turbines. Some waste heat recovery boilers are designed to permit generation of additional steam by burning conventional fuels in a firebox to supplement the heat from the hot exhaust gases. Such systems are referred to as supplementary-fired combined cycle gas turbines.

3.2.2 Emissions from Stationary Gas Turbines

The pollutants emitted from gas turbines are those common to all combustion sources, NO_x, HC, CO, SO_2, particulates, and visible emissions. The mass emissions from stationary gas turbines will differ depending on several

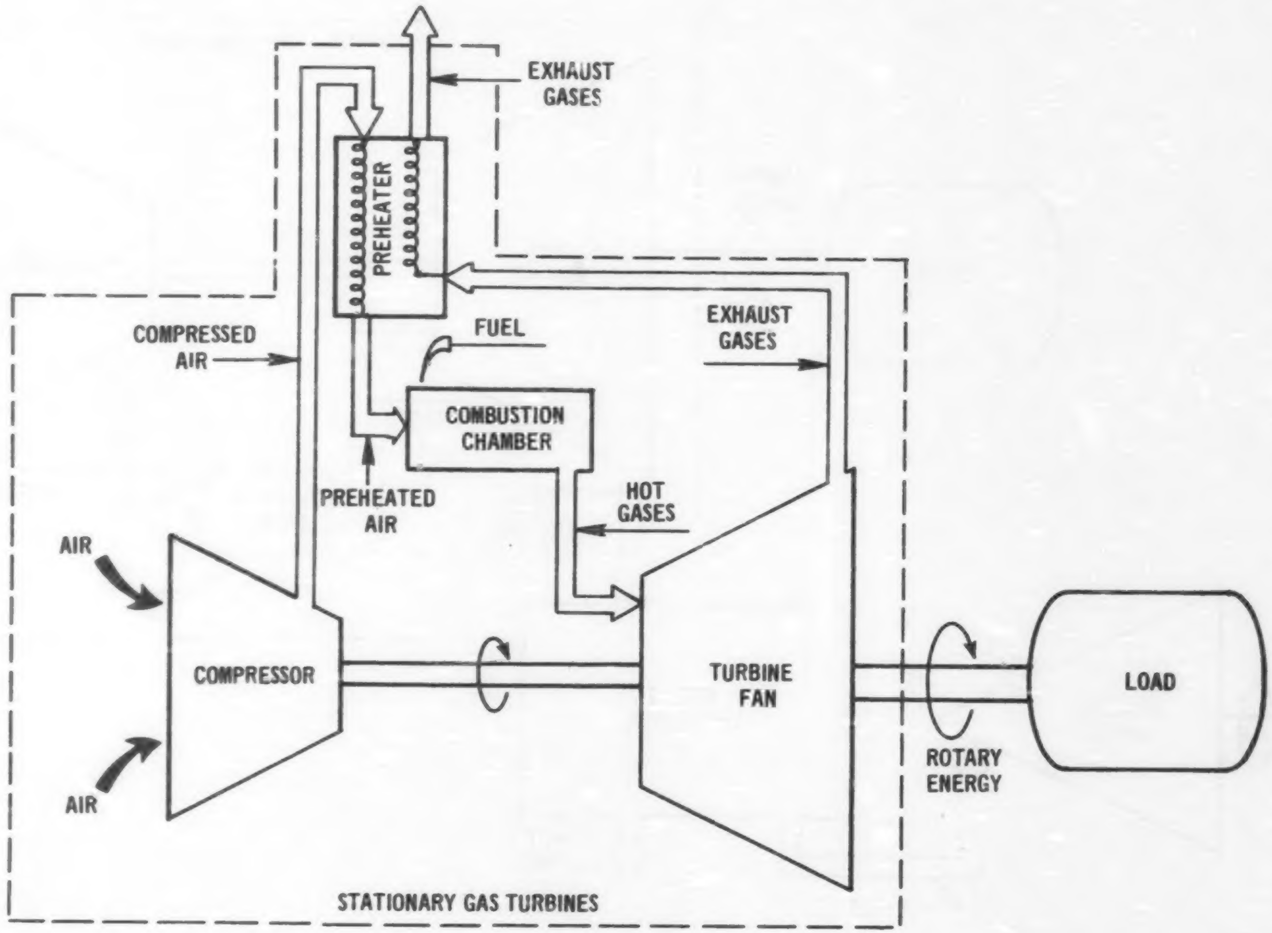

EXHAUST GASES
PREHEATER
COMPRESSED AIR
FUEL
EXHAUST GASES
COMBUSTION CHAMBER
PREHEATED AIR
HOT GASES
AIR
AIR
COMPRESSOR
TURBINE FAN
LOAD
ROTARY ENERGY
STATIONARY GAS TURBINES

Figure 3-9. Typical regenerative cycle gas turbine.

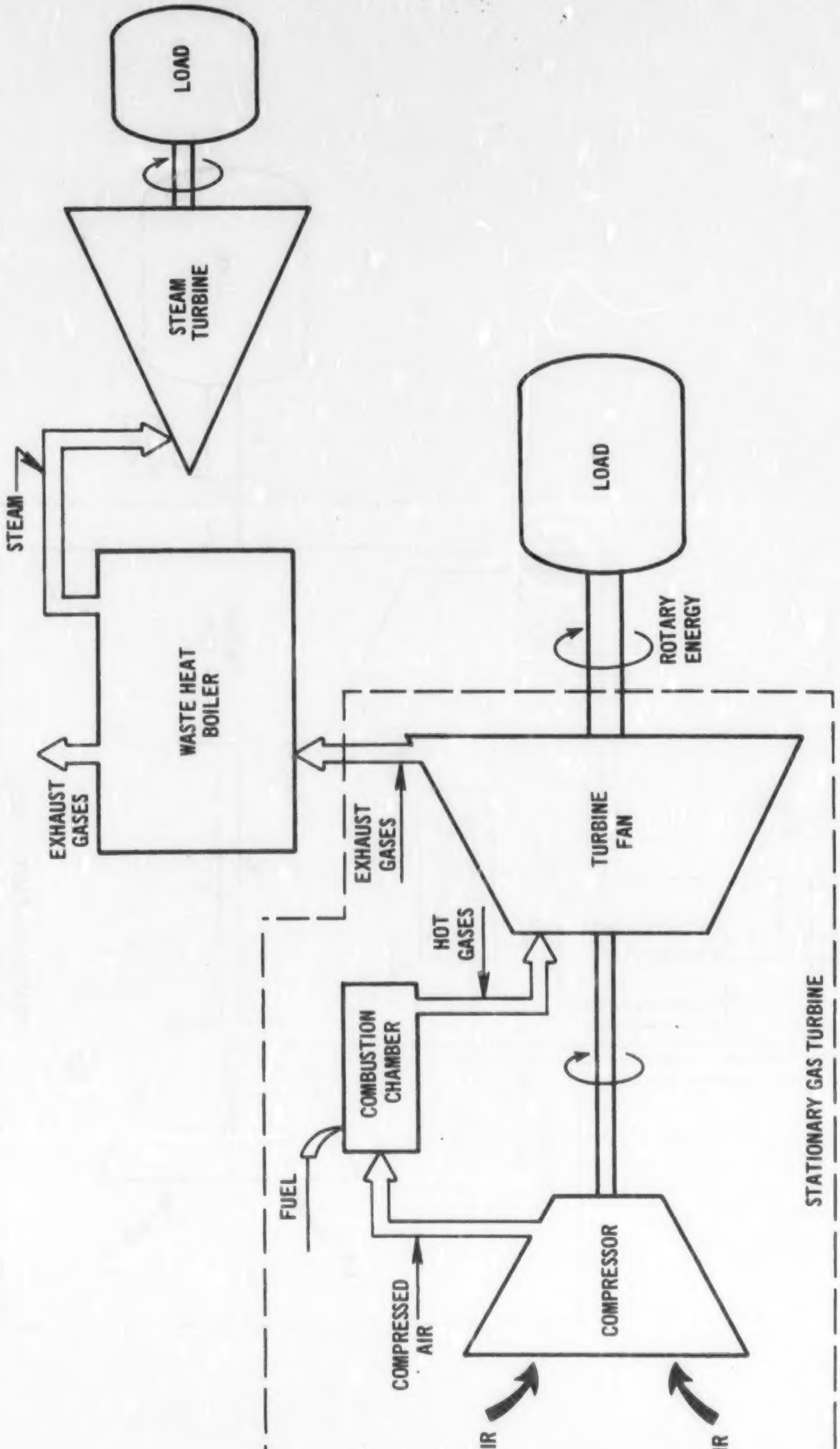

LOAD
STEAM TURBINE
STEAM
WASTE HEAT BOILER
EXHAUST GASES
LOAD
ROTARY ENERGY
EXHAUST GASES
TURBINE FAN
HOT GASES
COMBUSTION CHAMBER
FUEL
COMPRESSED AIR
COMPRESSOR
AIR
AIR
STATIONARY GAS TURBINE

Figure 3-10. Typical combined cycle gas turbine.

variables which are discussed in greater detail in the following section. Examples of some of the variables are: turbine firing temperature, turbine pressure ratio, turbine load, combustor design, and atmospheric conditions.

3.2.2.1 Simple Cycle and Combined Cycle Gas Turbines - Mass emissions from identical gas turbines used in the simple cycle and combined cycle (unfired boiler) configurations are equivalent and, therefore, only emissions from simple cycle machines will be discussed. If the combined cycle gas turbine uses a boiler with supplementary firing, the total emissions to atmosphere will be increased by pollutants generated in the boiler. Table 3-8 shows typical visible, CO, SO_2, and NO_x emissions for turbines of various sizes manufacturered by several companies when fired with both natural gas and liquid fuels. The table shows that for turbines operating at base load visible emissions are generally less than 10 percent opacity, NO_x emissions range from 0.94 lb/hr to 1578 lb/hr for turbines having outputs of 0.16 MW and 87.8 MW, respectively, and CO emissions range from 0 lb/hr to 47.2 lb/hr for turbines having outputs of 60 MW and 51.7 MW, respectively.

Sulfur dioxide emissions are a function of the efficiency of the gas turbine and the sulfur content of the fuel, since virtually all fuel sulfur is converted to oxide.[7, 8, 9] Gas turbines fired with premium low-sulfur fuels have low SO_2 emissions. A 33 MW gas turbine will produce about 36 pounds of SO_2 per hour when operated on 0.1 percent sulfur fuel, but emissions from this same turbine will increase to 690 pounds per hour when fired with 1.9 percent sulfur fuel. Table 3-8 shows that for turbines of various sizes, manufactured by several companies and fired with liquid fuels ranging from 0.01 percent to 0.5 percent sulfur by weight, SO_2 emissions range from 0 lb/hr to 540 lb/hr for turbines having outputs of 0.8 MW and 88 MW, respectively.

TABLE 3.8

VISIBLE, CO, NO$_x$, SO$_2$ EMISSIONS FROM STATIONARY GAS TURBINES PRODUCED BY SIX MANUFACTURERS

urbine	Manufacturer & Engine Model	Cycle Type	Fuel	Rated Base Load Power Output (MW)	Rated Heat Rate (Btu/KW-HR)	Plume Opacity (%)	NO$_x$ ppm @ 15%	NO$_x$ #/HR	NO$_x$ #/MWH	NO$_x$ G/sec	CO ppm @ 15%	CO #/HR	CO #/MWH	CO G/sec	Fuel Sulfur (WT %)	SO$_2$ #/HR	SO$_2$ #/MWH	SO$_2$ G/sec	DATA SOURCES
	GARRETT-AIRESEARCH																		
1	GTC 85-90	S.C.	JET A	0.158 (0.150)	23,854	< 10	55	0.94	6.25	0.12	321	3.3	22.18	0.42	0.01 0.1	0.039 0.39	0.26 2.59	0.005 0.05	AIRESEARCH letter dated 1/23/76, J. Massis to E. Noble Garrett Stationary G.T. Emission Control Program Status report dated 1/30/76, p. 3-58
2	831 - 800		N.G.	0.51 (0.46)	15,742	< 10	109 149	3.69 4.76	7.37 10.27	0.47 0.60	7 21	0.15 0.41	0.29 0.89	0.02 0.05	0 0.01 0.1	0 0.09 0.88	0 0.17 0.71	0 0.01 0.11	
	SOLAR																		
3	Saturn	S.C.	N.G. Kerosene	0.8	15,690	<10 ≃10	51 69	2.64 4.03	3.30 5.06	0.33 0.51	68 163	2.13 5.76	2.65 7.20	0.27 0.73	0 0.01 0.1	0 0.17 1.71	0 0.14 1.36	0 0.02 0.17	Solar letter dated 8/26/74, R. Kress to D. Goodwin 1974 sawyers G.T. catalog, p. 28 (GSC 1200 & GSC 3000) Telephone call on 3/26/76, E. Noble to A. Finkelstein
4	Centaur	S.C.	N.G. Kerosene	2.5	13,947	< 10 >10	89 86	12.48 13.75	4.99 5.50	1.57 1.73	68 126	5.79 12.23	2.32 4.89	0.73 1.54	0 0.01 0.1	0 0.38 3.79	0 0.15 1.52	0 0.05 0.48	
	DETROIT DIESEL ALLISON																		
5	404 - 3	R.C.	N.G. DF-2	0.20	11,300	2 2	113 209	1.61 2.98	6.15 11.40	2.02 0.38	70 149	0.60 1.30	2.30 4.97	0.08 0.16	0 0.1 0.3 0.5	0 0.25 0.74 1.23	0 1.23 3.69 6.14	0 0.03 0.09 0.15	GM Report dated 3/21/73 from Automotive Emission Control Environmental Activities Staff to EPA/OAQPS/ESED
6 7	501K 13 501K 15	S.C.	N.G. DF-1	2.5	12,930	1 7	99 125	16.69 20.67	5.74 7.10	2.10 2.60	11 57	1.17 5.77	0.40 1.98	0.15 0.73	0 0.1 0.3 0.5	0 3.51 10.54 17.57	0 1.41 4.22 7.03	0 0.44 1.33 2.21	
	GENERAL ELECTRIC CO.																		
8	MS5001N/P	S.C.	DF-2	23.5 (22.2)	11,964	< 10	147	187	8.4	23.5	7.8	6.0	0.27	0.75	0.1 0.3 0.5	30.6 91.7 153	1.30 3.90 6.50	3.86 11.55 19.25	GE Letter dated 10/31/75, E. Zeltmann to K. Durkee " Page 55, Test Point 63
9	MS7001B	S.C. & CCUF	N.G. DF-2	60.3 (49) & (59.4) 80.8	10,846	< 10	119 163	263 371	5.37 5.78	33.4 46.7	2.3 0	3.1 0	0.06 0	0.39 0	0 0.1 0.3 0.5	0 71.1 213 355	0 1.18 3.54 5.89	0 8.96 26.87 44.8	" Page 58, Test Point 3A " Page 61, Test Point 15
10	MS7001B	R.C.	DF-2	57.4 (53)	8,876	< 10	373	930	16.2	117.8	5	7.6	0.16	0.97	0.1 0.3 0.5	55.4 166 277	0.96 2.89 4.82	6.98 20.93 34.9	" Page 62, Test Point 2
11	MS7001C	S.C. & CCUF	DF-2	67.4 (66.2) & 94.5	10,949	< 10	164	531	8.0	66.9	2.1	4.0	0.06	0.51	0.1 0.3 0.5	80.2 241 401	1.19 3.57 5.95	10.11 30.32 50.53	" Page 63, Test Point 8 GE paper. "Control of G.T. Emissions in the world environment."
	WESTINGHOUSE																		
12	1916	S.C.	DF-2	12.5	15,680	---	120	102	8.1	12.79	29	15.3	1.2	1.93	0.1 0.3 0.5	21.30 63.91 107	1.70 5.11 8.52	2.68 8.06 13.42	ASME report 72-JPG-GT-2 1974 sawyers G.T. Catalog Westinghouse letter dated 1/8/76, C. DeForse to D. Goodwin Telephone calls on 2/4/76 and 4/21/76, E. Noble to P. Holden Report GASL TR 787, dated 3/74, for NY Power Pool
13	W251B	S.C.	DF-2	29.2 (32.8)	13,420 (12,960)	---	189	390	11.9	49.2	31	39.4	1.2	4.97	0.1 0.3 0.5	42.59 128 213	1.46 4.38 7.29	5.37 16.10 26.83	
14	W501B4	S.C.	DF-2	78.85	11,320	< 20	227	925	11.69	116.5	17.5	43.9	0.66	5.53	0.1 0.3 0.5	97.0 291 485	1.23 3.69 6.15	12.22 36.67 61.12	
15	W501D	S.C.	DF-2	87.77	11,320	---	337	1578	17.97	199	---	---	---	---	0.1 0.3 0.5	108 324 540	1.23 3.69 6.15	13.61 40.83 68.0	
	TURBODYNE																		
16	11C	S.C.	DF-2	51.7	11,859	8	315	310	15.67	102.1	30	47.2	0.9	5.95	0.1 0.3 0.5	66.6 200 332	1.29 3.87 6.45	8.40 25.19 41.93	Report dated 12/19/75, from Turbodyne to D. Goodwin Telephone call on 2/5/76, E. Noble to N. Gaylord

TABLE 3-8B (Continued)

NOTES

A. NO_x and CO emissions were calculated using one of the following formulas:

1) $ER = 1.552 \times 10^{-7}$ (Ec) MQ

where: ER = pollutant emission rate (#/hr)

Ec = " concentration (ppm), dry basis (not corrected to 15% O_2)

M = pollutant molecular weight

NO_x (as NO_2) = 46

CO = 28

Q = stack flow (SCFM), at 29.9"Hg and 70°F

2) $ER = E_pP_h$

where: ER = pollutant emission rate (#/hr)

E_p = " mass " " (#/1000 hp-hr)

P_h = power output (hp)

3) $ER = E_h (H_f) W_f$

where: E_h = pollutant mass emissions rate (#/million Btu)

H_f = lower heating value of fuel (assumed 18,400 Btu/#)

W_f = fuel usage rate (#/hr)

B. Heat rates, when not provided directly, were calculated as follows:

1) $HR = SFC (H_f)$

where: HR = Heat input required per unit of work $\left(\frac{\text{Btu}}{\text{kw-hr}}\right)$

SFC = specific fuel consumption (#/kw-hr)

H_f = lower heating value of fuel (Btu/#)

2) $$HR = \frac{W_fH_f}{P_k}$$

where: W_f = fuel usage (#/hr)

H_f = lower heating value of fuel (Btu/#)

P_k = power output (kw)

TABLE 3-8B (Continued)

C. Sulfur to SO_2 conversion is assumed to be 100%, using the following formula:

$$ER = \frac{(M_{SO_2})}{(M_s)} HR \frac{(S)}{(H_f)} P_k$$

where: ER = SO_2 emission rate (#/hr)
M = molecular weight
S = fuel sulfur content (#S/#fuel)
HR = gas turbine heat rate (Btu/kwh)
P_k = power output (kw)

D. For the instances where emissions were measured at other than rated base load, the actual power output during the tests is indicated by the presence of brackets.

E. 1) S.C. refers to a simple cycle gas turbine
2) R.C. refers to a regenerative cycle gas turbine
3) CCUF refers to a combined cycle plant with an unfired heat. recovery system generator.
4) N.G. refers to natural gas
5) DF-1 refers to distillate fuel #1
6) DF-2 refers to distillate fuel #2

Unburned hydrocarbon emissions from gas turbines are due to vaporized unburned fuel or partially burned products which escape the combustion reaction zone and are emitted in the exhaust. Since combustion efficiencies of gas turbines operating at base load normally exceed 99 percent, hydrocarbon emissions at base load are not substantial and may range from 1 to 5 ppm total hydrocarbons measured as methane or hexane (1 ppm is approximately equal to 3 lb/hr for a 30 MW turbine).[11, 12, 13, 14] Ambient hydrocarbon levels during many of these tests for hydrocarbon emissions were also measured as 1 - 5 ppm.[15]

Particulate emissions from gas turbines consist of ash from the fuel and particulates of carbon and hydrocarbons resulting from incomplete combustion. Depending on the size of the turbine and the fuel burned, particulate emissions range from 1 lb/hr to over 40 lb/hr and vary from 0.002 gr/scf to 0.10 gr/scf.[16, 17, 18, 19] For example, particulate emissions from a turbine operating at 20 MW and burning natural gas were 4.8 lb/hr while those from a turbine operating at 52 MW and burning #2 oil were about 41 lb/hr. Somewhat higher levels of particulate emissions may result from gas turbines fired with crude or residual fuels, although some data indicate that particulate emissions when burning these fuels are actually less. When considering these data on particulate emissions, one should remember that the high flow velocities, extreme gas turbulence from the ducting configuration and sound baffles and the difficulty of sampling with very long probes in the environment of the exhaust gases results in particulate measurements of questionable accuracy and high variability.[21, 22, 23]

Visible emissions or smoke emissions from gas turbines may be caused by only a small portion of the total particulate emissions since they are comprised of the extremely small and finely dividied particulate matter.

Various studies have indicated that the major contributor to visible emissions are the particulates consisting of low density agglomerates of carbon which are generally less than 1 micron in size and of the same order of magnitude as the wave length of visible light thus producing a greater visual effect than larger particulates.[24, 25] Table 3-8 shows that visible emissions from stationary gas turbines operating at base load are generally less than 10 percent opacity.

3.2.2.2 Regenerative Cycle Gas Turbines - The kinds of pollutants emitted by regenerative cycle gas turbines are identical to those listed in Section 3.2.2.1 for simply cycle and combined cycle turbines. Table 3-8 shows NO_x CO and SO_2 emissions for a General Electric MS 7001 B simple cycle machine and a MS 7001 B regenerative cycle machine. In general, the following observations can be made for the two machines when operating at equal outputs: (1) CO emission levels are essentially the same for both machines; (2) NO_x emissions from the regenerative machine are greater by more than a factor of two; (3) SO_2 emission levels are less from the regenerative machine and are proportional to the relative efficiencies of the two machines.[1/]

No data are available for particulate emissions from regenerative cycle gas turbines. Since particulate emissions occur from the fuel ash and from the carbon and hydrocarbon particles formed by incomplete combustion, the regenerative cycle machine should have lower particulate emissions than an equivalent simple cycle machine. This is a logical conclusion since the

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[1/] This follows from the fact that all sulfur in the fuel is converted to SO_2 and the higher efficiency regenerative cycle machine requires less fuel for an equivalent output.

regenerative machine uses less fuel (and, hence, less ash is formed) per unit of output and the combustion chambers for both machines are identical.

Hydrocarbon emissions from regenerative cycle machines are of the same magnitude as those from simple cycle machines.[26]

3.2.3 Parameters that Influence Emissions from Stationary Gas Turbines

The pollutants emitted from stationary gas turbines are CO, HC, particulates, SO_2, NO_x, and visible emissions (as detailed in Section 3.2.1). There are several factors which influence the amounts of these pollutants which are emitted. These factors are discussed, by pollutant, in the following sections. SO_2 emissions are strictly a function of the sulfur content of the fuel and, therefore, are not discussed further.

3.2.3.1 Operation of a Typical "Can" Combustor - The formation of pollutants occurs during the process of combustion. In order to understand the parameters that influence emissions from stationary gas turbines, one must understand the basic operation and functions of the combustor. A schematic of a conventional style "can type" combustor is presented in Figure 3-11.[27] (The general description of the combustion process which follows will also be applicable to an annular type combustor.) Fuel is commonly introduced to the combustor can by the use of one or more nozzles which may use air or pressure atomization. The combustion air, cooling air, and dilution air are supplied to the combustor from the discharge of the compressor. The mixture of combustion air and fuel is ignited by an automatically retracting spark plug as shown in Figure 3-11 and when multiple combustors are used, the flame is propagated between combustor cans by "crossfire tubes". The combustion of the fuel is completed in the "reaction zone" where gas temperatures exceed 3040°F.[28, 29, 30]

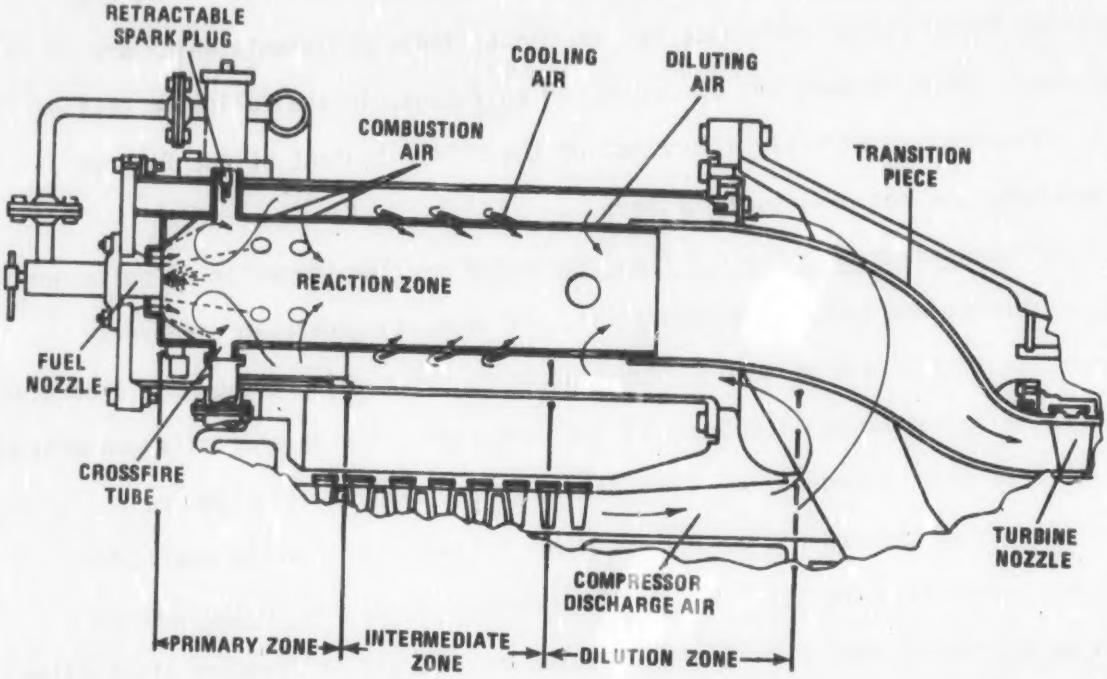

RETRACTABLE SPARK PLUG
COOLING AIR
DILUTING AIR
COMBUSTION AIR
TRANSITION PIECE
REACTION ZONE
FUEL NOZZLE
CROSSFIRE TUBE
TURBINE NOZZLE
COMPRESSOR DISCHARGE AIR
PRIMARY ZONE
INTERMEDIATE ZONE
DILUTION ZONE

Figure 3-11. Can type combustion system for gas turbines.[27]

The reaction zone consists of two subzones - the primary zone and the intermediate zone which are shown in Figure 3-11. The boundaries between zones are not well defined and reactions of one zone may extend into the subsequent zone.

Efficient combustion of the fuel requires proper fuel atomization, proper air to fuel ratios, and mixing, and sufficient time (residence time) within the reaction zone of the combustor for the chemical kinetics of the reaction between the fuel and oxygen to be completed. The combustion process is commonly stabilized in the primary zone of the combustor by aerodynamic recirculation of the combustion air entering the first row of holes in the combustor. The portion of air that flows upstream toward the fuel nozzle (See Figure 3-11) is preheated as it mixes with the products of combustion and thus provides continuous ignition of the fresh charges of fuel. This continuous ignition of injected fuel by the preheated recirculation gases is important to provide stable combustion over a wide range of combustor inlet conditions and air to fuel ratios.[31, 32, 33] Completion of the combustion reactions occurs in the intermediate zone. Cooling air is introduced in the intermediate zone to provide a boundary layer along the inside wall of the combustion chamber and to remove heat from the wall by convection. Dilution air, admitted in the dilution zone, then mixes with the products of combustion to reduce gas temperatures to the design turbine inlet temperature which can approach 1950°F in present day machines operating at base load.[34]

Satisfactory combustor operation requires attainment of the following objectives (not considering the reduction of air pollutants):[35, 36]

a. efficient combustion of the fuel (or fuels)

b. low pressure drops across the combustor (since any pressure drops are parasitic and result in reduced cycle efficiencies)

c. stability of operation over a wide range of fuel to air ratios (since fuel to air ratio varies proportionately with variations in turbine load, this ratio will therefore change over a wide range each time the turbine is started and taken to full load operation)

d. dependable ignition

e. low component maintenance and long life

f. a controlled temperature distribution at the turbine inlet such that temperature profiles do not exist which will result in failure of the turbine buckets.

3.2.3.2 Factors Which Affect Particulate and Visible Emissions from Stationary Gas Turbines -

3.2.3.2.1 Particulate emissions - Particulate emissions consist of ash carbonaceous agglomerates and HC. The ash content is primarily a function of the fuel. Fuels containing high ash and vanadium contents will result in higher particulate emission rates than light distillate fuels. Also, inhibitors added to the fuel to retard vanadium corrosion of the turbine components will increase particulate emissions.[37, 38]

Under some state and local regulations, unburned hydrocarbons (discussed in Section 3.2.2.3) are considered as particulate matter if they are exhausted as liquid droplets or allowed to condense in the sampling train.[39] However, this should not add a significant amount to the emission rate except at turbine start-up and at low turbine loads.[40, 41] All these particulate emissions contribute to particulate loading.

3.2.3.2.2 Visible emissions - Smoke - Although literature references are somewhat contradictory, it is generally agreed that the major contribution to visible emissions result from gas turbines when particles of carbonaceous material formed during the combustion process agglomerate to sizes that are in the range of the wavelength of visible light (0.45 to 0.65 micrometers) and form smoke. These carbonaceous particles are caused by the incomplete combustion of hydrocarbon fuels and the thermal cracking of fuel in hot regions of the combustor with locally insufficient oxygen.[42, 43, 44] A plume discoloration can occur under certain circumstances due to conversion of the emissions of NO to NO_2. Factors which affect visible emissions are discussed below.

3.2.3.2.2.1 Fuels - Some fuels may form more smoke or visible emissions than others if compensatory design changes are not incorporated in the combustion system. Generally speaking, visible emissions caused by these carbonaceous materials are not experienced when gaseous fuels are used as shown in Figure 3-12, which shows a smoke spot reading of zero for a turbine burning natural gas (smoke spot number increases with increasing visible emissions). For liquid fuels, the paraffinic saturated fuels tend to "smoke" less than the aromatic or unsaturated fuels and this smoking tendency is related to the chemical bond energies necessary to completely consume the fuel.[46, 47]

Fuel hydrogen content and residual carbon content also affect visible emissions. A reduction in hydrogen content or an increase in residual carbon, or both, can increase visible emissions.[48] Figure 3-13 shows the effect of fuel hydrogen content on smoke for various turbine operating loads. Figure 3-14 shows the effect of fuel hydrogen content on smoke for two designs of gas turbine combustors, one specifically designed to reduce visible emissions. Figure 3-15 presents visible emissions versus the firing temperature of the gas turbine when burning a typical east coast residual fuel. All these

Figure 3-12.[45] Visible Emissions vs. Load for a Turbine Using Various Fuels

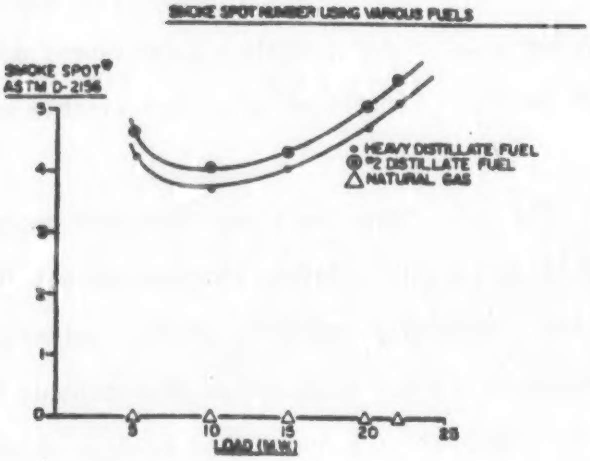


Figure 3-13.[49] Visible Emissions vs. Load for Two Fuels with Different Hydrogen Contents (GE MS-7001 and MS-9001 Turbines)

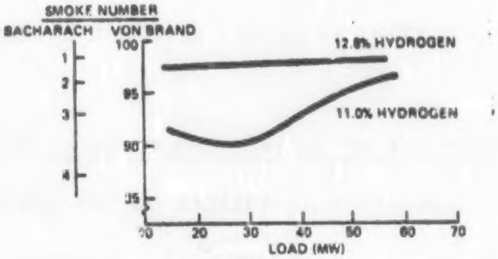


Figure 3-14.[50] Visible Emissions vs. Fuel Hydrogen Content for Two Combustor Designs (GE MS-7001B and MS-9001 Turbines)

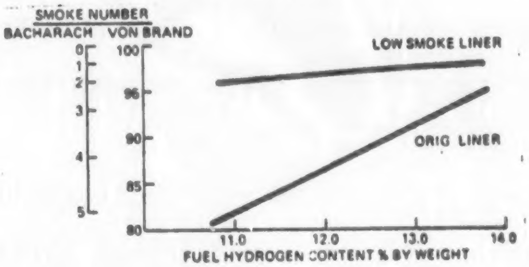
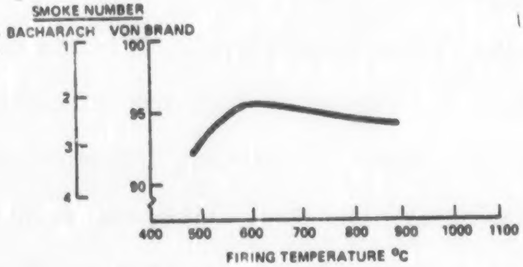


Figure 3-15.[51] Visible Emissions vs. Firing Temperature (Load) for a GE MS-5001 Turbine Burning Residual Fuel



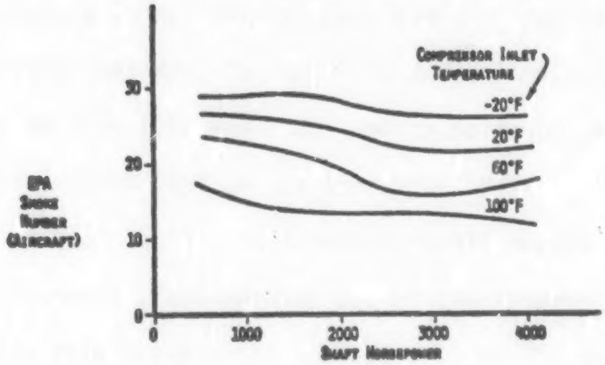
NOTE: Visible emissions decrease with increasing Von Brand number and decreasing Bacharach number.

turbines (Figures 3-13, 3-14, and 3-15) were equipped with combustors to reduce visible emissions except for the one labeled "orig. liner" in Figure 3-14. With the one exception, emissions were not in the visible range.[52]

3.2.3.2.2.2 The effect of ambient temperatures on visible emissions - The effect of ambient inlet temperature variations on the visible emissions from gas turbines is shown for one engine in Figure 3-16. Reported results of the effect of inlet temperature on visible emissions have been contradictory but the experience of at least one other manufacturer indicates a slight increase in smoke with decreased temperature.[54] This increase is believed due to the increased quenching effects of the main combustion air and cooling air at the combustor side walls.

3.2.3.2.2.3 Other factors - The other factors which affect visible emissions are combustor design parameters which primarily relate to the fuel to air ratios and mixing of the fuel and air.[55, 56, 57, 58] Carbon can be formed in the primary zone of the combustor if local areas of fuel rich mixtures exist. Combustor operating pressure can affect visible emissions as can be observed from Figures 3-12, 3-13, and 3-14 (since combustor pressures vary with operating load and firing temperature). Possible explanations for the effect of pressure level are changes in fuel droplet evaporation temperature with pressure, differences in oxidant diffusion rates with pressure and changes in the spray parameters of the combustion chamber fuel nozzle with pressure.[59] It is generally agreed, however, that the most important considerations for reducing visible emissions involve designs which provide leaner fuel to air ratios (less fuel; more air), minimize the number of localized fuel rich areas, provide effective fuel atomization and mixing

Figure 3-16.[53] Visible Emissions vs. Shaft Horsepower for Various Compressor Inlet Temperatures (GM, 501K Industrial Turbine, Constant Speed, JP-5 Fuel)



NOTE: Visible emissions increase with increasing EPA smoke number.

with air and result in sufficient lean regions within the combustor for smoke burnout.[59, 60, 61, 62, 63]

3.2.3.3 Factors Which Affect Emissions of Hydrocarbons and Carbon Monoxide -

3.2.3.3.1 Effect of combustion efficiency and operation at low turbine loads - Incomplete combustion is the principal cause of emissions of hydrocarbons (HC) and carbon monoxide (CO).[64, 65, 66] Since gas turbine are typically designed for optimum efficiency at full load, they normally operate in excess of 99 percent combustion efficiency at full load. For gas turbines used in aircraft applications, this efficiency typically drops to the 90 to 95 percent range for operation at idle or low power conditions.[67, 68] A similar efficiency drop occurs in gas turbines used for stationary applications. Because of this drop in efficiency, emissions of HC and CO from the turbines will be higher for turbine start-up and operation at low loads and will be a minimum at full load operations. Figure 3-17 shows HC emissions versus fuel to air ratio, and Figure 3-18 shows CO emissions versus firing temperature. Since fuel to air ratio and firing temperature are proportional to gas turbine load, Figures 3-17 and 3-18 demonstrate the trend for HC and CO emissions to increase with decreasing turbine load - becoming an exponential increase at low loads.

The low combustion efficiency (and, hence, higher HC and CO emissions) experienced in gas turbines operating at low loads is due to the poor burning conditions.[71, 72, 73, 74] Low combustion inlet air temperatures cause quenching to occur thus terminating combustion before completion, and the low fuel to air ratios (fuel lean combustion) result in low equivalence ratios [1/] in the

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[1/] Equivalence ratio (Ø) is defined as the ratio of the actual fuel to air ratio to the stoichiometric fuel to air ratio. For "fuel rich" combustion (excess fuel) Ø exceeds 1.0 while for "fuel lean" combustion (excess air), Ø is less than 1.0.

Figure 3-17 [69] Hydrocarbon Emissions vs. Fuel-to-Air Ratio (Load) for the GE MS-7001B and MS-5001N Turbines When Burning Oil Fuel

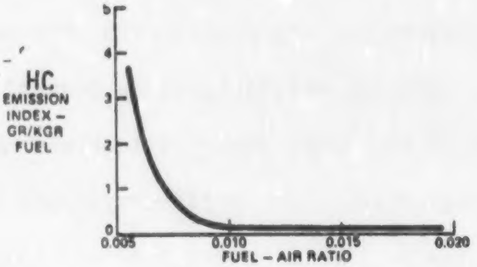
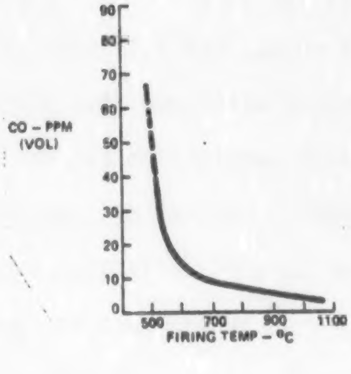


Figure 3-18 [70] Carbon Monoxide Emissions vs. Firing Temperatures (load) for GE Gas Turbines Burning Oil Fuel



primary zone of the combustor thus reducing burning intensity. The low fuel and air flows at low power operation result in poor fuel atomization and distribution.

Figure 3-19 shows the relationships between idle power combustion efficiency and CO and HC emissions and represents data from tests performed on several aircraft-type gas turbines manufactured by General Electric. The specific ratio of HC and CO emissions from gas turbines operating at low power conditions (and, therefore, lower efficiency) also varies with specific design features of the engine combustor and fuel injection systems. As shown in Figure 3-19, the ratio of CO to HC emissions increases as the efficiency level increases. This is consistent with the chemical kinetics of combustion reactions which show that HC compounds are consumed faster than CO with the result that as gas turbine efficiency is increased, any remaining non-equilibrium products of combustion will tend to exist mainly as CO.[76,77] The data from tests of these same engines also show that the levels of CO and HC emissions from gas turbines with high compression ratios (ratio of compressor inlet air pressure to compressor outlet or combustor inlet pressure) are less than CO and HC emissions from turbines with lower compression ratios. This relationship is demonstrated in Figure 3-20, and one can conclude that the increased combustor inlet temperatures and pressures due to higher engine compression ratios generally result in improved combustion efficiency at idle or low power conditions.

3.2.3.3.2 Effects of ambient conditions - Figure 3-21 demonstrates the effect of variations in ambient temperatures on CO and HC emissions from the 501K industrial gas turbine produced by Detroit Diesel Allison, Division of General Motors. This figure shows that increases in ambient temperature result

Figure 3-19[75] Relationships Between Combustion Efficiency and Levels of CO and HC Emissions (Data corrected to ISO conditions)

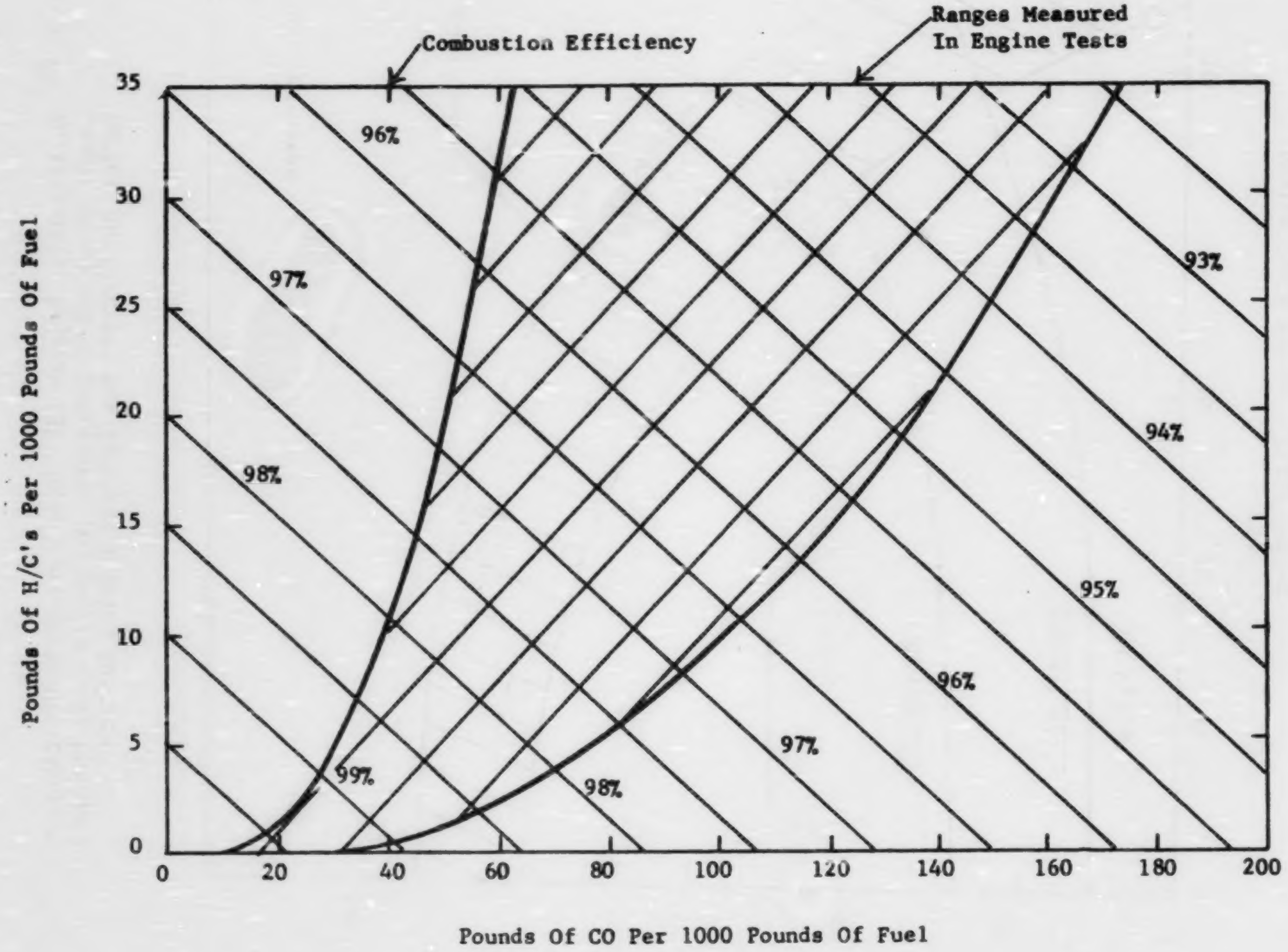


Figure 3-20 [78] Relationships Between Idle Power Combustion Efficiency Levels of CO and HC Emissions, and Engine Cycle Pressure Ratios for Aircraft Gas Turbines Burning Kerosene

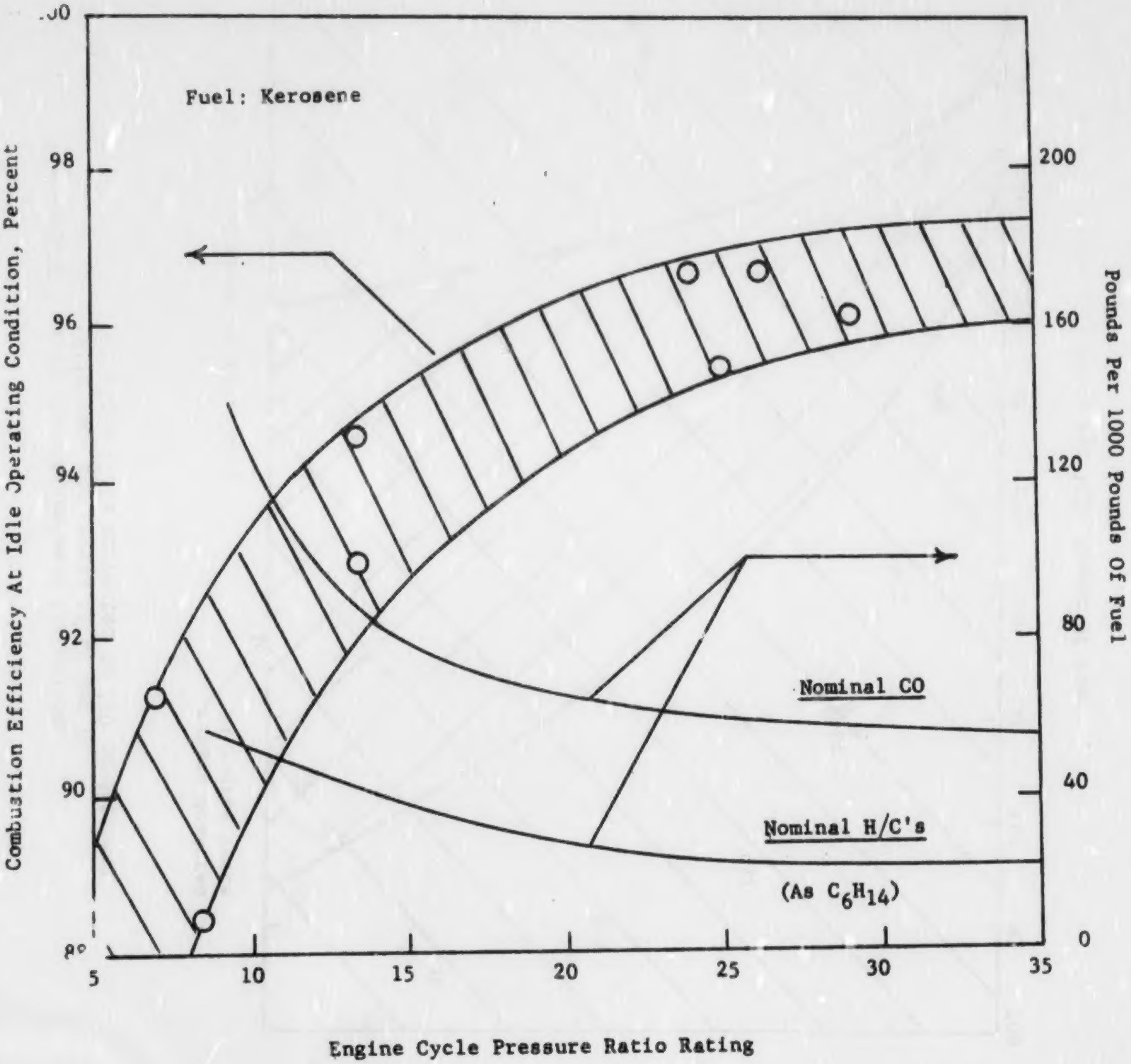


Figure 3-21 [79] Effect of Inlet Temperature on CO and HC Emissions (501-K Industrial Turbine, Constant Speed, JP-5 Fuel)

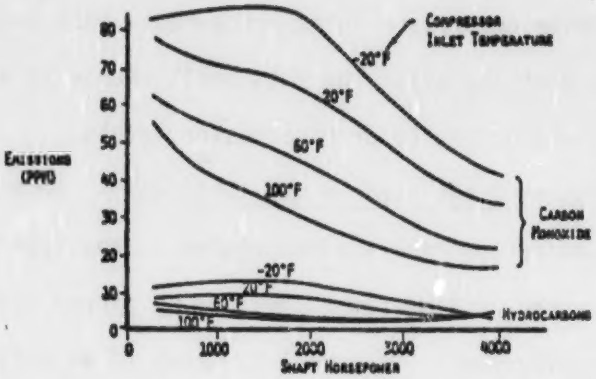
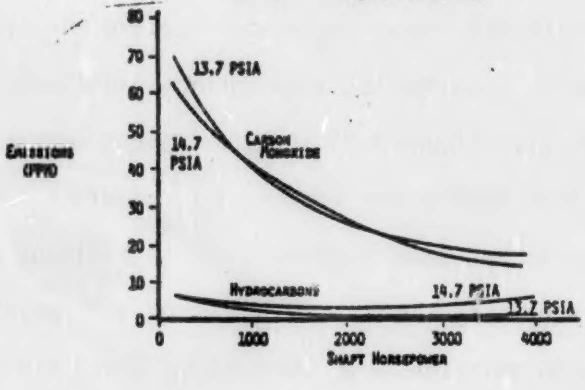


Figure 3-22 [80] Effect of Inlet Pressure on CO and HC Emissions (501-K Industrial Turbines, Constant RPM, JP-5 Fuel, 100°F Compressor Inlet Temperature)



in decreases in HC and CO emissions over the engine load range. The effects of variations in inlet temperature will also vary depending on engine and combustor design so the data of Figure 3-21 cannot be extrapolated to other engine designs although they do indicate the general trends.

Figure 3-22 shows CO and HC emissions versus turbine load for the 501K engine for two difference compressor inlet pressures - 13.7 psia and 14.7 psia. This figure indicates that the effect of this small change in ambient pressure did not significantly change the CO or HC emission levels.

3.2.3.3.3 Effects of fuels used in gas turbines - Another factor which may affect HC and CO emissions from a given engine is the type of fuel burned. These effects can be somewhat alleviated by proper design of the combustor to burn specific fuels. Figure 3-23 shows CO emissions versus load for a 26 MW gas turbine produced by Westinghouse Electric Corporation when burning a heavy distillate fuel, No. 2 distillate fuel, and natural gas. The data do not indicate any significant differences (small differences indicated could be attributed to data scatter) in the levels of CO emissions when different fuels are burned although the CO emissions when burning heavy distillate fuel are consistently above those when burning the other fuels. Figure 3-24 shows HC versus load for the same turbines and fuels and again no consistent trend is evident. Figure 3-25 shows emissions versus load for the Detroit Diesel Allison 501K engine when burning two different fuels. The data indicate a slightly higher level but again not significant increase of CO and HC emissions when burning the heavier No. 2 oil and operating at full load. At low load, the CO emissions when burning No. 2 oil are significantly higher (about a factor or two) than the emissions when burning JP-5.

Figure 3-23[81] CO Emissions vs. Turbine Load for Various Fuels

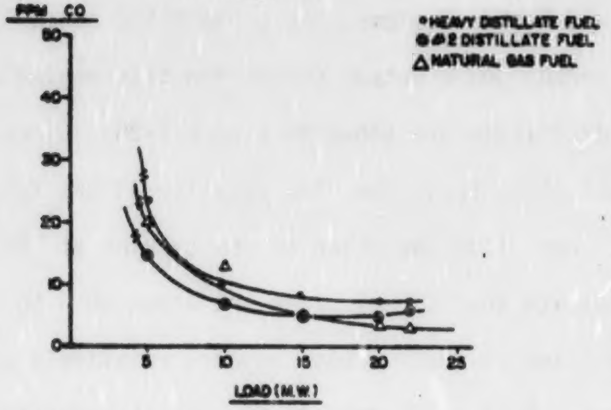


Figure 3-24[82] HC_6 Emissions vs. Turbine Load for Various Fuels

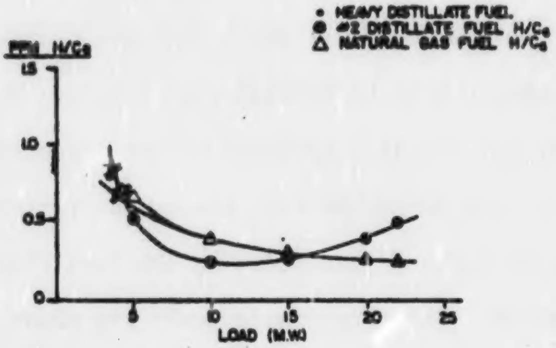
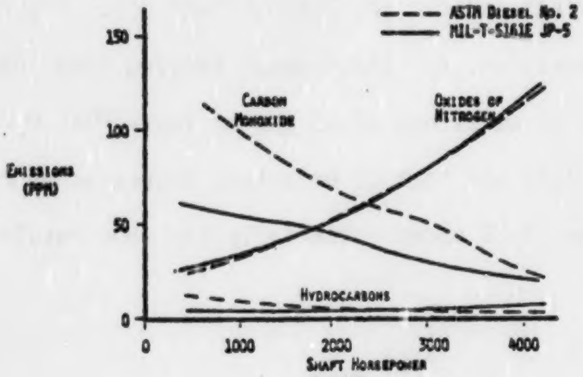


Figure 3-25[83] Emissions vs. Load for a Turbine Burning Two Different Fuels



Some experiments have been performed when using methanol as a gas turbine fuel. Turbo Power and Marine Systems, Inc., ran tests on an FT4C-1 gas turbine and the CO versus power output curves for this engine when burning methanol and No. 2 distillate are shown in Figure 3-26. Since this engine was designed to burn distillate fuel, the flow capacity of the fuel system when burning methanol did not allow operation of the turbine at its rated load of 29 MW. The data indicate that CO emissions are about 40 - 50 ppm greater when burning methanol and this difference appears relatively constant over the 8 to 20 MW load range. Data on HC emissions were not reported but since CO emissions at the higher loads were low, one would expect HC emissions to also be low since their reaction time is less than that of CO.

Coal gas is an emerging fuel which has potential use for firing in gas turbine combustors. Pillsbury and Lin of Westinghouse Electric Corporation have obtained emission data from a full scale coal gas combustor test rig (for use in the Westinghouse W 501D) turbine when burning synthetic coal gas with various heating values and at a combustor inlet temperature of 450°K (350°F).[85] These tests were performed with the combustion rig initially simulating mid-load operation and then reducing the fuel flow rate to simulate the idle mode of operation. Data from these tests are shown in Figures 3-27 and 3-28, which show CO emissions versus combustor exit temperature (which is proportional to load) for synthetic coal gases with heating values varying from 200 Btu/scf to 90 Btu/scf and for No. 2 distillate oil. Figure 3-27 shows that, at mid-load operation, all three gases ranging from 105 Btu/scf to 200 Btu/scf burned with CO emissions of 50 ppm or less, that emissions increased with decreasing load and that CO emissions increased with decreased fuel heating value. Figure 3-28 shows essentially the same results for the

Figure 3-26[84] CO Emissions vs. Power Output for a Gas Turbine Burning Methanol and No. 2 Distillate Oil

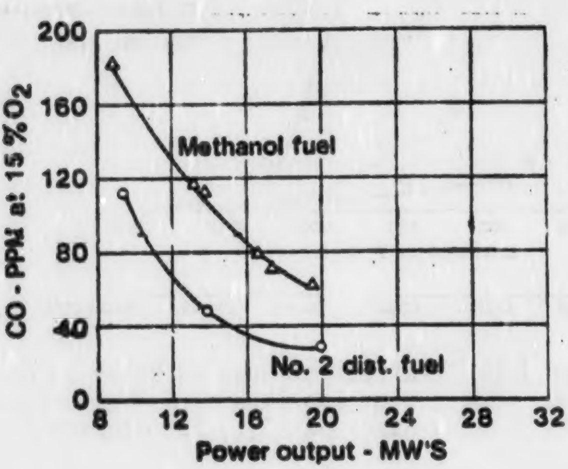


Figure 3-27[86] Effect of Heating Value on CO Emissions from Gas Turbine Combustors Burning Coal Gases of 105 to 200 Btu/scf

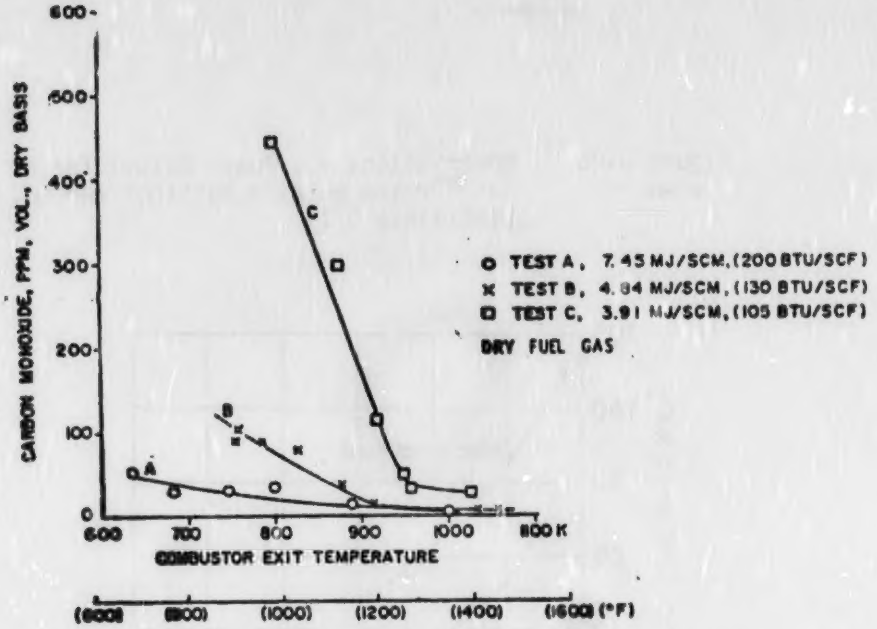
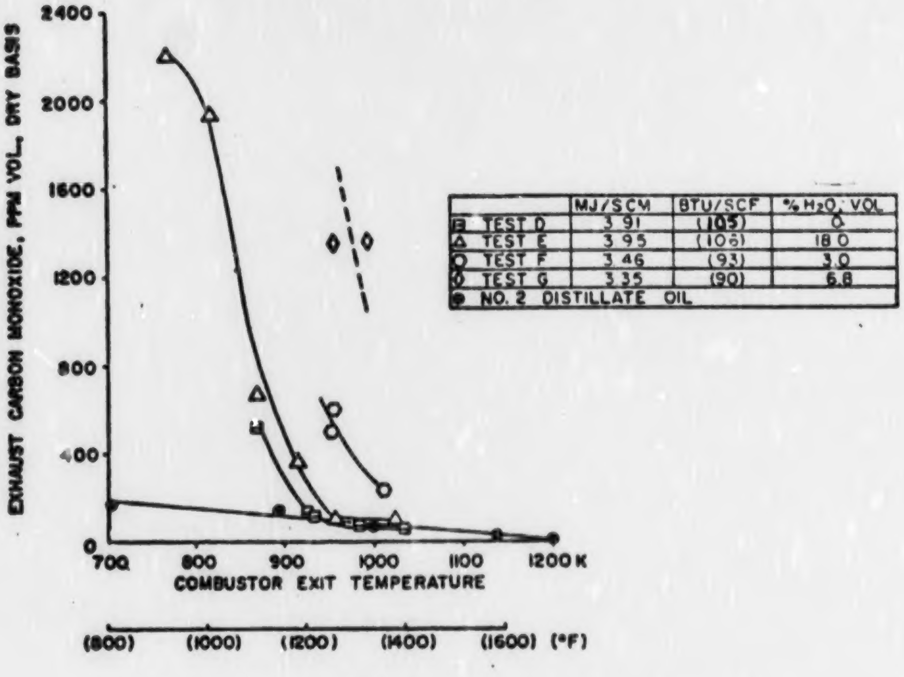


Figure 3-28[87] Effect of Heating Value on CO Emissions from Gas Turbine Combustors Burning Coal Gases of 90 to 105 Btu/scf



105 and 106 Btu/scf gases as that shown for Test C in Figure 3-27. Tests F and G as shown in Figure 3-28 show a rapid increase in CO emissions as gas heating value drops. Additional test data (F & G) were not available because of combustor flameout at lower firing temperatures. These tests also showed that the burner combustion stability was poor after the point where the CO emissions for each test started to rise rapidly. For comparison, the CO emissions from this combustor when fired with No. 2 fuel oil are also shown on Figure 3-28.

3.2.3.4 Factors Which Affect Emissions of Nitrogen Oxides (NO_x) - Nitrogen oxides produced by combustion of fuels in stationary gas turbines are formed by the combination of nitrogen and oxygen in the combustion air ("thermal" NO_x) and from the combination of nitrogen in the fuel with oxygen from the combustion air ("organic NO_x).

3.2.3.4.1 Thermal NO_x and the variables effecting its formation - The NO_x formation mechanism by the combination of atmospheric nitrogen and oxygen (thermal NO_x) for fuel lean combustion in the combustor primary zone is commonly described and modeled using the Zeldovich mechanism which is described by the following equations:[88,89,90,91]

a. $O_2 \rightleftharpoons 2O$

b. $N_2 \rightleftharpoons 2N$

c. $N + O \rightleftharpoons NO$

d. $N + O_2 \rightleftharpoons NO + O$

e. $O + N_2 \rightleftharpoons NO + N$

The reactions were expanded by others to include an intermediate hydroxide molecule (OH) in the fuel rich region:[92,93]

f. $N + OH \rightleftharpoons NO + H$

and to include the intermediate product of hydrogen cyanide (HCN) by combination of a nitrogen atom with a hydrocarbon molecule in the fuel rich combustion zone.[94,95]

g. $N_2 + CH \rightleftharpoons HCN + N$

h. $N + OH \rightleftharpoons H + NO$

These reaction mechanisms are commonly used by the manufacturers of gas turbines in computor models to predict emissions from their conventional combustors which operate at, or near, stoichiometric combustion conditions.[96,97] They are not valid for combustors which may be developed to operate at off-stoichiometric conditions (such as fuel rich combustion) and thereby result in reduced NO_x emissions.

The Zeldovich equation for the formation rate of thermal nitric oxide (ppm/m sec) is:[98]

$$\text{i.} \quad \frac{d\, X_{NO}}{dt} = k\, P^{0.5}\, X_{O_2}{}^{0.5}\, X_{N_2}\, T_f{}^{-1}\, e^{-122{,}000/T_f} \left[1 - \frac{X_{NO}}{(X_{NO_e})^2}\right]$$

where:

K = constant of formation

P = combustor pressure

T_f = flame temperature

X = the volumetric fraction of the substance (i.e., O_2; N_2; NO)

(X_{NO_e}) = nitric oxide equilibrium concentration

Equation (i) shows that the thermal nitric oxide formation rate is extremely sensitive to the flame temperature -- increasing exponentially with increases in flame temperature. From this equation, we can also observe that the formation of NO varies with the square root function of the combustion pressure and with the N_2, O_2, and NO concentrations. As the concentrations of nitric oxide reaches its equilibrium value, the bracketed figure (equation i) becomes zero and thus the nitric oxide formation rate becomes zero.

The thermochemical equilibrium quantities of NO that can be generated depend strongly on the combustion temperature levels and on the availability of oxygen. Thus, the equilibrium quantities of NO produced increase rapidly with increases in combustor inlet temperature and as the fuel to air equivalence ratio approaches values of 0.8 to 1.0. Figure 3-29 illustrates the thermochemical equilibrium quantities of NO_x which can be produced in combustion processes. Fortunately, the quantities of NO_x emissions generated by gas turbines are limited by the short residence times of the hot combustion gases within the engine combustors and by the rapid quenching of these gases with the dilution air. Therefore, very high combustion efficiencies can be attained without generating the thermochemical equilibrium quantities of NO.[100, 101, 102, 103] Some typical nitric oxide formation rate data for gas turbines are shown in Figure 3-30. Figures 3-30 and 3-31 demonstrate the tremendous effect of combustor inlet temperature on the formation rate of NO. Figure 3-30 also shows that NO formation rates increase as combustor inlet pressures increase and that operation of the combustion process at equivalence ratios less than or greater than stoichiometric (equivalence ratio = 1.0) will result in reduced formation of NO.

Humidity contained in the combustion air will effect NO_x emissions by reducing the combustion flame temperatures. Figure 3-32 illustrates the effect of humidity and ambient temperatures on NO_x emissions from one gas turbine. NO_x emissions are shown to increase with increasing ambient temperatures (0 percent relative humidity curve) and to decrease when humid air is used (100 percent relative humidity curve compared to the 0 percent relative humidity curve).

Figure 3-29[99] Thermochemical Equilibrium Levels for NO_x Emissions

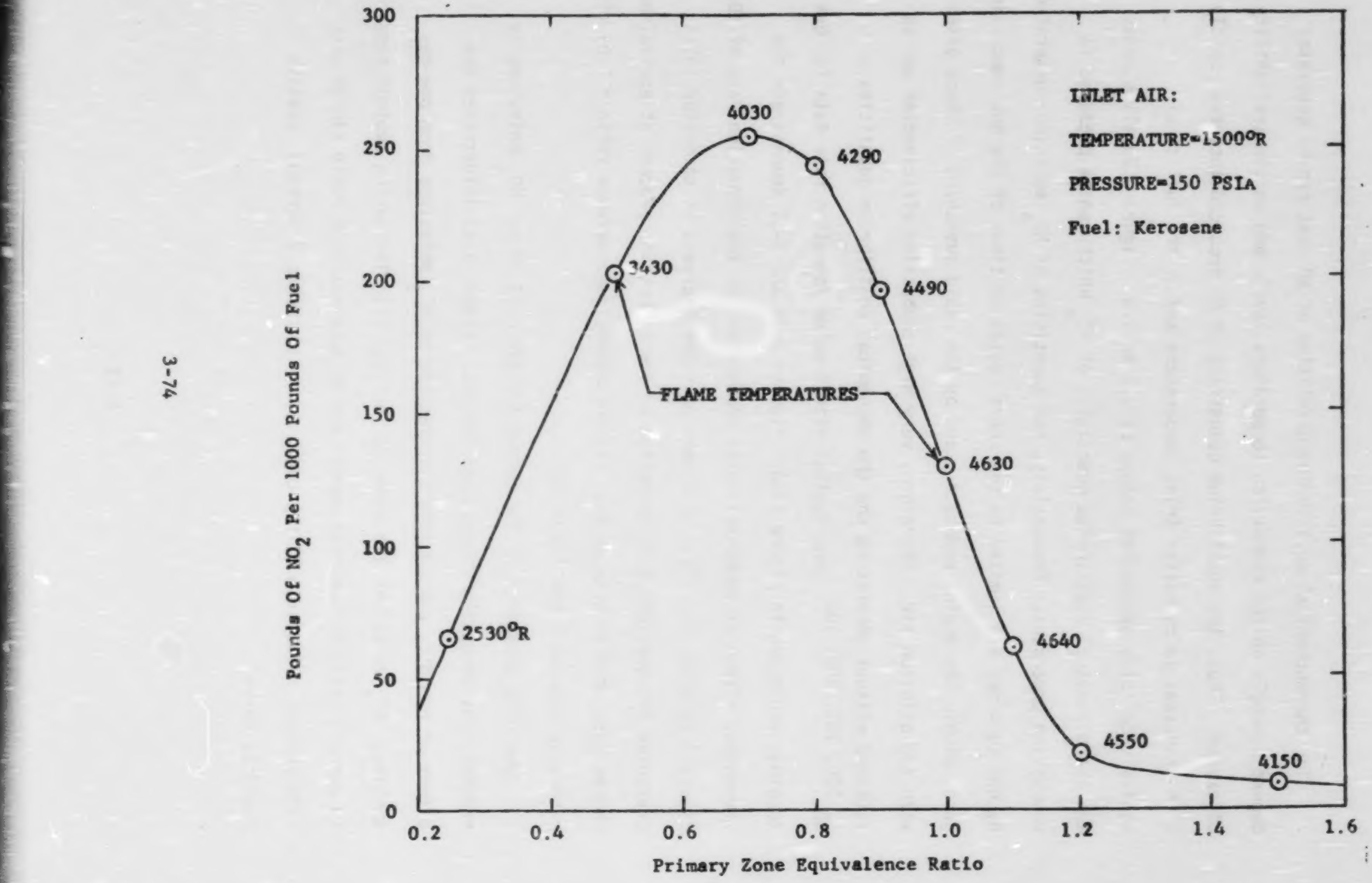


Figure 3-30[103] Formation Rate Data for NO_x

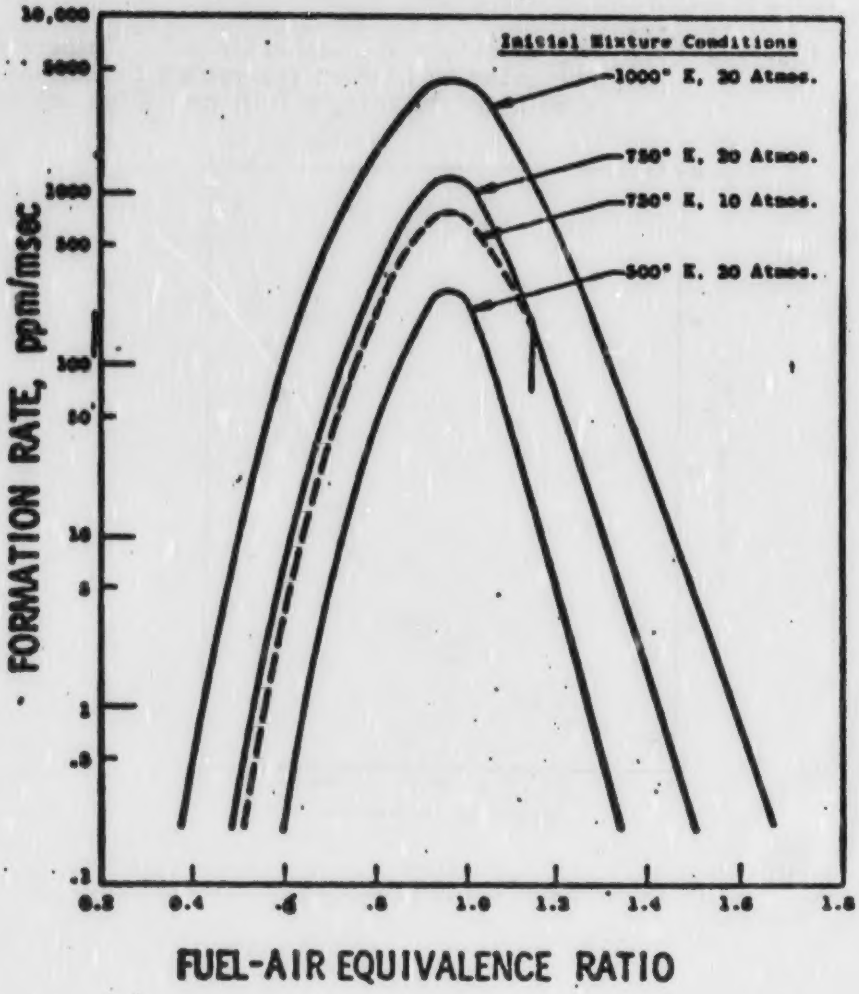


Figure 3-31[104] NO_x emissions vs. Combustor Inlet Temperature as Presented by Lipfert (corrected to ambient temperature = 80°F and humidity = 0.01 gm H_2O/gm dry air)

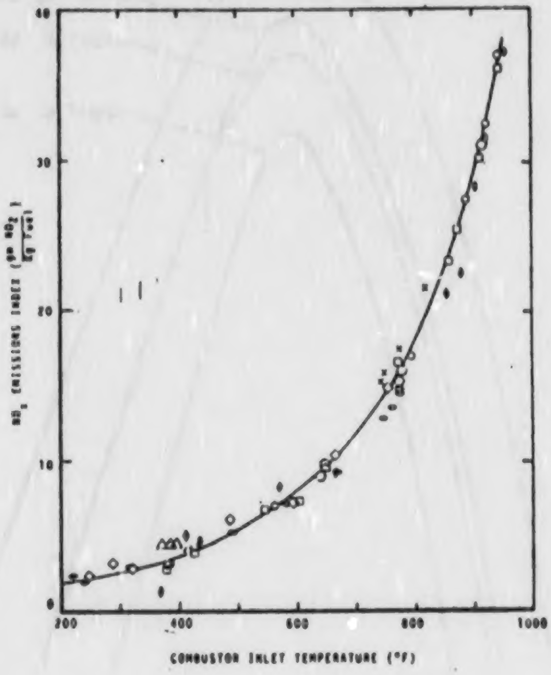


Figure 3-32[105] The Effect of Ambient Temperature and Humidity on NO_x Emissions from a Turbine

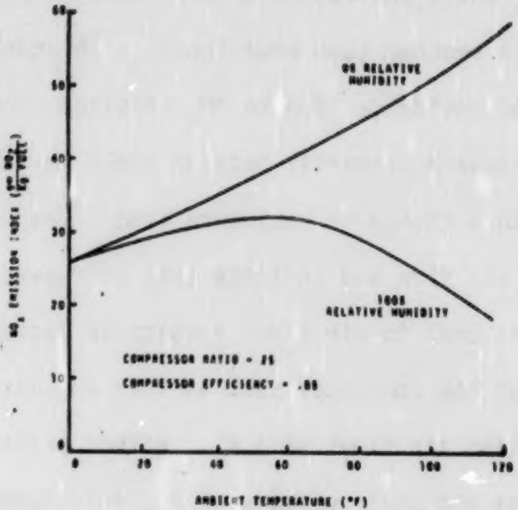
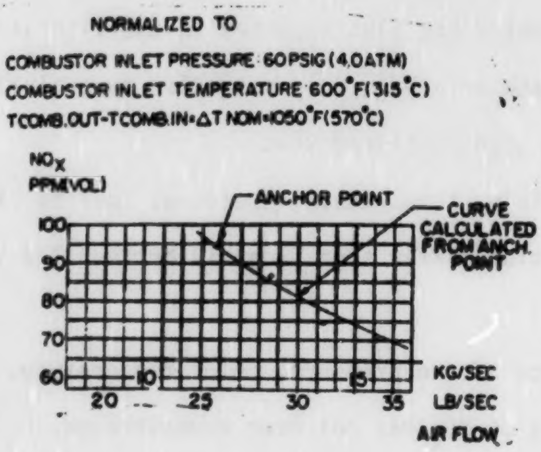


Figure 3-33[106] Influence of Residence Time on NO_x Emissions



Increased residence time of the gases in the combustor will result in increased emissions of NO_x until the residence time becomes so great that the NO equilibrium value is reached (see equation 1 - bracketed quantity). Data on the exact effect of residence time on NO_x emissions from a given gas turbine combustor are scarce primarily because other variables (such as mass flowrates) change with a change in residence time. The data in Figure 3-33 show NO_x emissions versus air flow and indicate that increased residence time (which is inversely proportional to air flow) results in increased NO_x emissions.

The specific design of the combustor such as fuel and air injection methods and combustor cooling air flows will also effect emissions. Discussion of some of these variations and their effects on NO_x emissions can be found in Chapter 4 of this document.

There have been many attempts to develop correlations between NO_x emissions and the many variables (such as temperature, pressure, humidity, residence time, air to fuel ratio, etc.) which effect these emissions. Sullivan and Mas prepared a paper in July 1975 titled, "A Critical Review of NO_x Correlations for Gas Turbine Combustors" in which they reviewed the semi-empirical and empirical correlations which had been reported in the literature.[107] They concluded that an adequate data base does not exist to evaluate the various NO_x correlations. They also concluded that:

a. The effect of inlet temperature, pressure, fuel to air ratio and mass flow on NO_x production differ significantly between the various correlations.

b. The independence of the pressure, inlet temperature, fuel to air ratio and airflow effect on NO_x has not been demonstrated.

During the process of developing standards of performance for stationary gas turbines, several correlations were provided to EPA and these are summarized in Table 3-9. Equations 1 and 2 in Table 3-9 were submitted with the recommendation that they be used to correct emissions from gas turbines, regardless of manufacturer, to a common set of reference conditions. Manufacturers of gas turbines were quick to respond that these equations were not applicable to their engines and stated that, in their opinion, individual equations would probably have to be developed for each gas turbine and combustor combination. Equations 3, 4 and 5 of Table 3-9 were submitted with additional data to show their applicability and accuracy when applied to engines of the specific manufacturer. Data presented by the manufacturers indicate that equation 3 can correct emissions to an accuracy of ± 10 percent, equation 4 has a reported accuracy of ± 7.5 percent, equation 5 and equations developed by Detroit Diesel Allison Division of General Motors also have been shown to predict emissions very accurately for specific machines.

Despite their obvious differences, the equations reported in the literature and those shown in Table 3-9 do have some common characteristics. These common characteristics are summarized as follows:

a. Most pressure dependence data supports the conclusion that the correction factor for the effect of pressure on NO_x emissions is of the form:[117, 118]

$$NO_x = NO_{x\ OBS} \left(\frac{P_3 \text{ reference}}{P_3 \text{ observed}} \right)^{0.5}$$

Where P_3 equals the compressor discharge or regenerator discharge pressure.

b. Most humidity dependence data supports the conclusion that the correction factor for the effect of humidity on NO_x emissions is of the form:

Table 3-9. NO_x Correlations for Gas Turbines

Source of Correlation	Equation	Definition of Terms
1. Combustion and Fuels Committee, American Society of Mechanical Engineers[108]	$NO_x = [(NO_x)_{obs}(\frac{P_{3ref}}{P_{3obs}})^{0.5} e^{19(H-.00633)} e^{c(T_{3ref} - T_{3obs})}]$ $C_e D(f/a\ ref - f/a\ obs)\ e^{(T_{3obs})(f/a\ obs) - T_{3ref}(f/a\ ref)]}$ $(\frac{21-15}{21-O_{2obs}})(\frac{21-O_{2obs}}{21-O_{2ISO}})]$	P_3 = Compressor discharge or regenerator discharge pressure H = specific humidity T_3 = compressor discharge or regenerator discharge temperature f/a = fuel/air ratio by weight O_2 = oxygen concentration by volume C, D, E = empirical constants obs = observed ISO = International Standard Organization, standard atmospheric conditions: 15°C (519°R), 1.013 Bar. (14.7psia), 60% relative humidity $(.00633 \frac{lb\ H_2O}{lb\ air})$ ref = reference O_{2ISO} = determined from commonly used gas turbine performance calculations or curves.
2. American National Standard Institute[109]	$NO_x = (NO_x)_{obs}[\frac{10}{Pobs}]^{0.5}[e^{0.005(315 - Tobs)}][e^{19(Hobs - .0064)}]$ $[\frac{20.9-15}{20.9-(O_2)_{obs}}] - 18640\ [\frac{(f/a)obs}{1 + (f/a)obs}]\ \%\ N$	P = burner inlet pressure in atmosphere T = burner inlet temperature in °C H = specific humidity ~ gm H_2O/gm air O_2 = exhaust oxygen, percent by volume f/a = fuel/air weight ratio N = fuel bound nitrogen ~ percent by weight obs = observed during tests Base conditions for equation P = 10 atmosphere (i.e. turbine has 10/1 compression ratio) T = 315°C (approximate heat of compression for 10 atm.) H = .00633 $\%O_2$ = 15 % base as suggested by EPA in previous draft documents %N = NO_2 fuel has essentially no bound N
3. General Electric[110] - "A Constant Residence time equation for Correcting NO_x Emissions to a Set of Gas Turbine Operating Conditions--"	$NO_x = NO_{x\ ref}\ (\alpha)(\frac{f/a}{f/a\,ref})(\frac{P_3}{P_{3ref}})^{0.5}(e^{-19\Delta H + C_1\Delta T_3 + C_2 \frac{\Delta f/a}{T_3}})$	NO_{xref} = NO_x measured at reference conditions of f/a, P_3, H, T_3. $\alpha = (\frac{NO_{xref}\ machine\ X}{NO_{xref}\ machine\ Y}) = 1$ for single machine model P_3 = compressor or regenerator discharge pressure T_3 = Compressor or regenerator discharge temperature H = specific humidity ~ wt H_2O/wt dry air Ref = reference condition f/a = fuel/air ratio by weight $\Delta H = H - Href$ $\Delta T_3 = T_3 - T\ ref$ $\frac{\Delta f/a}{T_3} = \frac{f/a}{T_3} - \frac{f/a\ ref}{T_{3ref}}$ C_1 = .00214 = empirical constant C_2 = 39,320 = empirical constant applies to constant speed simple cycle and regenerative machines and simple cycle variable speed machines.
4. Pratt & Whitney - Turbo Power and Marine[111]	$EI_{NOx} = EI_{NOxobs}[\frac{(Pb)\ ref}{(Pb)\ obs}]^{0.5}\ e^{0.00313[(Tbi)ref-(Tbi)obs)]}$ $(e^{19(H - .00634)})$	EI = emission index ~ $\frac{g\ NO_x\ emitted}{kg\ fuel}$ Pb = burner pressure Tbi = temperature at burner inlet H = specific humidity obs = observed conditions ref = reference conditions
5. Westinghouse[112]	$\frac{NO_x}{(NO_x)ref} = (\frac{P}{Pref})^{0.5}\ e^{[(\frac{T-Tref}{255}) + (\frac{\Delta T-\Delta Tref}{641}) + 12(H\ ref-H)]}$	P = compressor discharge pressure - Psia T = Compressor discharge temperature - °F ΔT = combustor temperature rise - °F H = specific humidity ref = values of parameters at the reference condition NO_x = ppm(v) - expressed as NO_2 $\Delta T \propto f/a$
6. Detroit Diesel Allison Division of General Motors[113]	The humidity correction factor used is $e^{19(H-.00633)}$ Other corrections were obtained by computer analysis of test data to provide equations for correlating emissions from the 501 K engine.	
7 Solar[114]	The equation which GE proposed for correcting NO_x emissions (see item 3 of this table) is not applicable to Solar engines. Solar believes the constants in that equation will differ "depending on whether the engine is single shaft, two shaft, simple cycle, or regenerative cycle as well as combustor type."	
8. Garrett Aireasearch[115]	Emission data not sufficient to indicate a preferable correlation. Use humidity correction: NO_x corrected = (NO_x measured) $e^{(19H - .00634)}$ Temperature correction: Lipfert curve (from literature) fits their engines fairly well.	
9. Turbodyne[116]	Humidity $NO_x = (NO_x\ measured)\ e^{15.73\ (H - .00634)}$ Temperature: "It appears that for each gas turbine there exists a unique relationship between ambient temperatures and NO_x emissions." They suggest that each manufacturer be permitted to establish the NO_x - temperature curve for each specific engine - combustor combination.	

$$NO_x = (NO_{x\ OBS})\ e^{K(Hobs - Href)}$$

Where K, a dimensionless constant, has been shown to be equal to 22 ± 8.[119] Review of the literature and the tabulation in Table 3-9 reveals that K=19 is the value most commonly chosen although some manufacturers (as shown in Table 3-9) have developed a specific constant for their machine.[120, 121] Ideally, the development of a specific constant for each machine is the better approach since it has been analytically calculated that the correction factor for humidity will vary with compression ratio, which is a function of machine design, as demonstrated in Figure 3-34.

c. Sullivan and Mas plotted NO_x emissions versus ambient temperature at constant combustor outlet temperature for the various correlations found in the literature. These data are shown in Figure 3-35 which shows that all the correlations except one predicted a small variation in NO_x due to ambient temperature changes. This relative insensitivity of NO_x emissions to ambient temperature variations can be explained by several compensating effects:[124]

(1) As the ambient temperature increases, combustor pressure and air to fuel ratio decrease, thus tending to decrease NO_x emissions.

(2) Concurrently with (1) above, the combustor inlet temperature increases and the air flowrate decreases thus tending to increase NO_x emissions.

Based on the preceding, one can conclude that the important ambient conditions which effect NO_x emissions from stationary gas turbines are the ambient pressure and humidity with ambient temperature having relatively little effect. This conclusion is not, however, substantiated by the data developed by Detroit Diesel Allison Division of General Motors Corporation as shown in Figure 3-36.

Figure 3-34[122] Correction Factors for Ambient Humidity

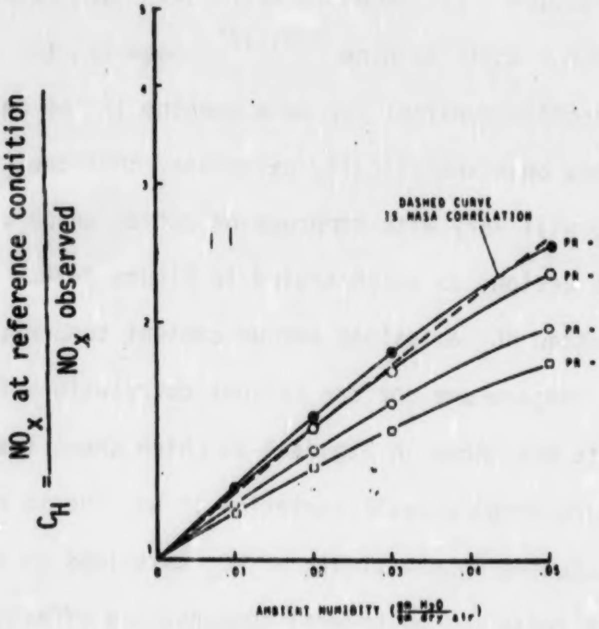


Figure 3-35[123] NO$_x$ Emissions vs. Ambient Temperature as Predicted by Various Correlations

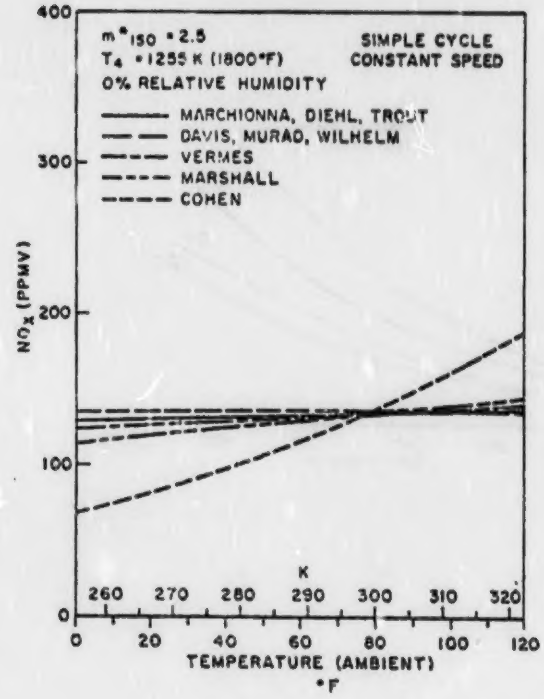
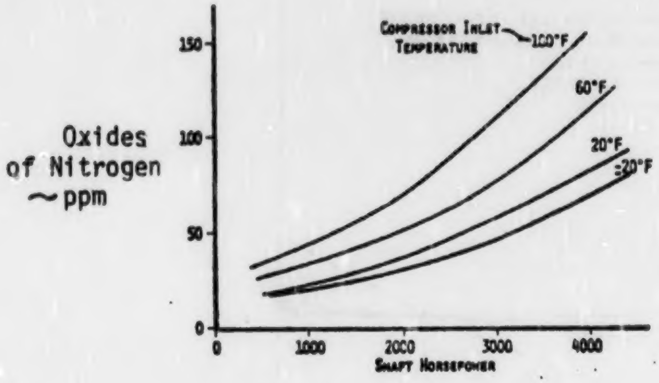


Figure 3-36[125] NO$_x$ vs. Shaft Horsepower for Various Compressor Inlet Temperatures (501-K Industrial Turbine, constant speed, JP-5 Fuel)



This figure shows that NO_x emissions about double when the compressor inlet temperature is varied from $-20^{\circ}F$ to $100^{\circ}F$ changing from approximately 70 ppm to 150 ppm at an output of 4000 horsepower. Based on this, the different corrections for temperature reported in the literature and Table 3-9 and comments of manufacturers it is concluded that the specific design of the gas turbine will affect the nature of the NO_x ambient temperature relationship and, therefore, a unique relationship will exist for each turbine model.[126,127,128,129] A correction factor for the effects of ambient temperature on NO_x emissions which is universally applicable to all gas turbines does, therefore, not exist.

Also, it must be remembered that the correlations shown in Table 3-9 and commonly reported in the literature were developed for combustors which operate at stoichiometric conditions in the primary zone and, therefore, should not be applied without verification to low emissions combustors which commonly utilize some means of suppressing flame temperature below stoichiometric values. For example, NO_x emissions from combustors which operate with a fuel rich primary zone have been found to decrease with increases in fuel to air ratio while NO_x emissions from the more conventional combustors with lean primary zones have been found to increase with increased fuel to air ratio.[130,131] The slope of this increase for the conventional combustor depends on specific combustor design features and thus generalized fuel-air ratio correlation factors have not been developed.[132] For this same reason, when using correction factors for extrapolating NO_x emissions data acquired at one set of ambient conditions to any other ambient conditions such as ISO conditions, one should require that the turbine be operated at a reference air to fuel ratio or firing temperature (such as full load turbine design conditions) while emissions measurements are being made.[133]

3.2.3.4.2 Organic NO_x and variables affecting its formation - Organic NO_x is formed during combustion by the chemical combination of the nitrogen atoms contained in the fuel molecule with oxygen from the air.[134, 315, 136, 137] Although many papers have been written on the conversion of fuel bound nitrogen (organic nitrogen to NO_x), the exact mechanism of NO_x formation is not known[138,139,140] Generally, organic NO_x is only a problem when burning residual oils, blends, some crude oils, or heavy distillate fuels which have high nitrogen contents.[141,142] Most light distillate fuels have less than 0.015 percent by weight nitrogen, crude oils generally have less than 0.2 percent and residual oils can go as high as 2 percent.[143, 144]

The maximum organic NO_x that will be produced when burning a given fuel in a gas turbine, assuming all fuel bound nitrogen is converted, may be calculated from the following equation:[145]

$$\text{NO (ppm by volume)} = \frac{N_f \times W_f}{W_f + W_w + W_a} \times \frac{30}{14} \times \frac{28.95}{30} \times 10^6,$$

where N_f = fuel-bound nitrogen, percent by weight (expressed as a decimal

W_f = fuel flow rate, lbs/sec

W_a = combustor air flow rate, lbs/sec

W_w = water injection flow rate, lbs/sec

30 = molecular weight of NO

28.95 = average molecular weight of combustor exhaust gas

14 = atomic weight of nitrogen

This simplifies to:

$$NO = \frac{N_f \times W_f/W_a \times 2.0679 \times 10^6}{(1 + W_f/W_a + W_w/W_a)} \qquad \text{(ppm by volume)}$$

In their experiments with a combustor from the MS 5000 series gas turbine, General Electric Corporation found that the actual conversion of fuel bound nitrogen varied with the nitrogen content of the fuel and to a lesser extent with the fuel to air ratio at which the gas turbine was operated. They therefore performed experiments using the MS 5000 series combustor operating at a constant fuel to air ratio of 0.021 to develop the curve shown in Figure 3-37. This curve shows that the percentage of fuel bound nitrogen converted to NO_x decreases as the fuel nitrogen content increases. Using this curve and the equation given above for calculating the NO_x contribution assuming 100 percent conversion of the fuel nitrogen, one can then calculate the NO_x contribution of the fuel to the total NO_x emissions from the MS 5000 gas turbine operating at a 0.021 fuel to air ratio.

The general applicability of the curve shown in Figure 3-37 to combustors with different designs has not been shown. For example, data developed by Dilmore of Westinghouse shows that for a fuel with 0.79 percent bound nitrogen the yield factor was about 20 to 30 percent compared to the 40 to 50 percent shown for the G.E. machine in Figure 3-37 (both machines were operated at firing temperatures in the 1800°F range).[147] The data in Dilmore's report (Reference 147) also show that the percent conversion of fuel bound nitrogen decreases with increased firing temperature while G.E. data indicates that the percent conversion increases with increasing firing temperature.[148] It has also been reported that the yield of NO_x from bound nitrogen in the fuel is less for fuel rich combustion than for combustion with leaner fuel mixtures.[149,150] As pointed out earlier in this section, the exact mechanism of NO_x formation from fuel bound nitrogen is not known. The variables which affect the emission of organic NO_x are not defined and further research is needed in this area.

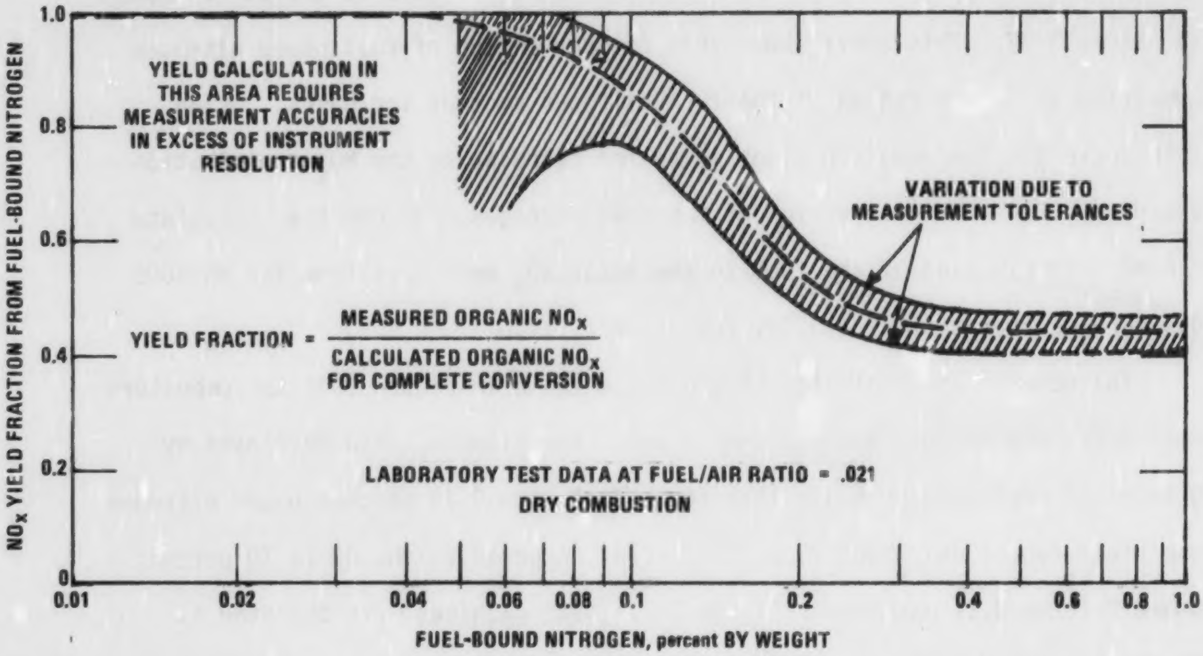


Figure 3-37[146]. NO_x yield fraction vs. fuel-bound nitrogen content.

3.2.3.4.3 The effects on NO_x emissions of burning various fuels in stationary gas turbines - The effect of fuel bound nitrogen on NO_x emissions from gas turbines has been discussed in section 3.2.3.4.2 of this report and, therefore, will be not discussed again.

It has been known for some time that burning natural gas instead of number 2 distillate in a given gas turbine can result in reducing NO_x emissions by as much as 50 percent.[151] This has been theorized to occur because the combustion of liquid fuel droplets proceeds in the reaction zone at fuel to air ratios which result in localized peak flame temperatures (which are conducive to greater NO_x formation) while the combustion of natural gas occurs at lower, more uniform temperatures.[152,153] The relative levels of NO_x emissions from one gas turbine firing natural gas, heavy distillate ($<$ 0.05 percent nitrogen) and number 2 distillate ($<$ 0.05 percent nitrogen) are shown in Figure 3-38. No significant difference in NO_x emissions for the heavy or number 2 distillate fuels was observed.

Figure 3-39 shows NO_x emissions versus firing temperature for a simple cycle gas turbine produced by General Electric and burning crude and distillate oils. Again, no significant difference in NO_x emissions is observed. No other data for NO_x emissions from gas turbines burning crude oils are available.

Estimated emissions from a FT4C-1D gas turbine manufactured by Turbo Power and Marine Systems when burning number 2 distillate and methanol are shown in Figure 3-40. The NO_x emissions are reduced by about 75 percent when methanol fuel is burned rather than the number 2 distillate. Data supplied by General Electric Corporation from tests performed on a production combustor for their MS 5000P gas turbine show a similar trend, indicating a 63 percent reduction in emissions of NO_x when burning methanol as compared to number 2 distillate.[157]

Figure 3-38[154] NO_x Emissions vs. Load for a Turbine Burning Various Fuels

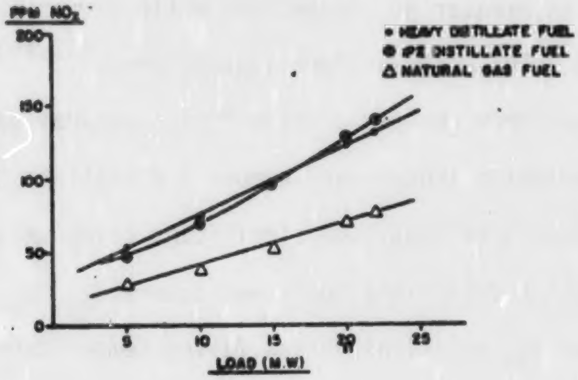


Figure 3-39[155]

NO_x Emissions vs. Firing Temperature for a GE MS7001 Turbine Burning Crude and Distillate Fuels

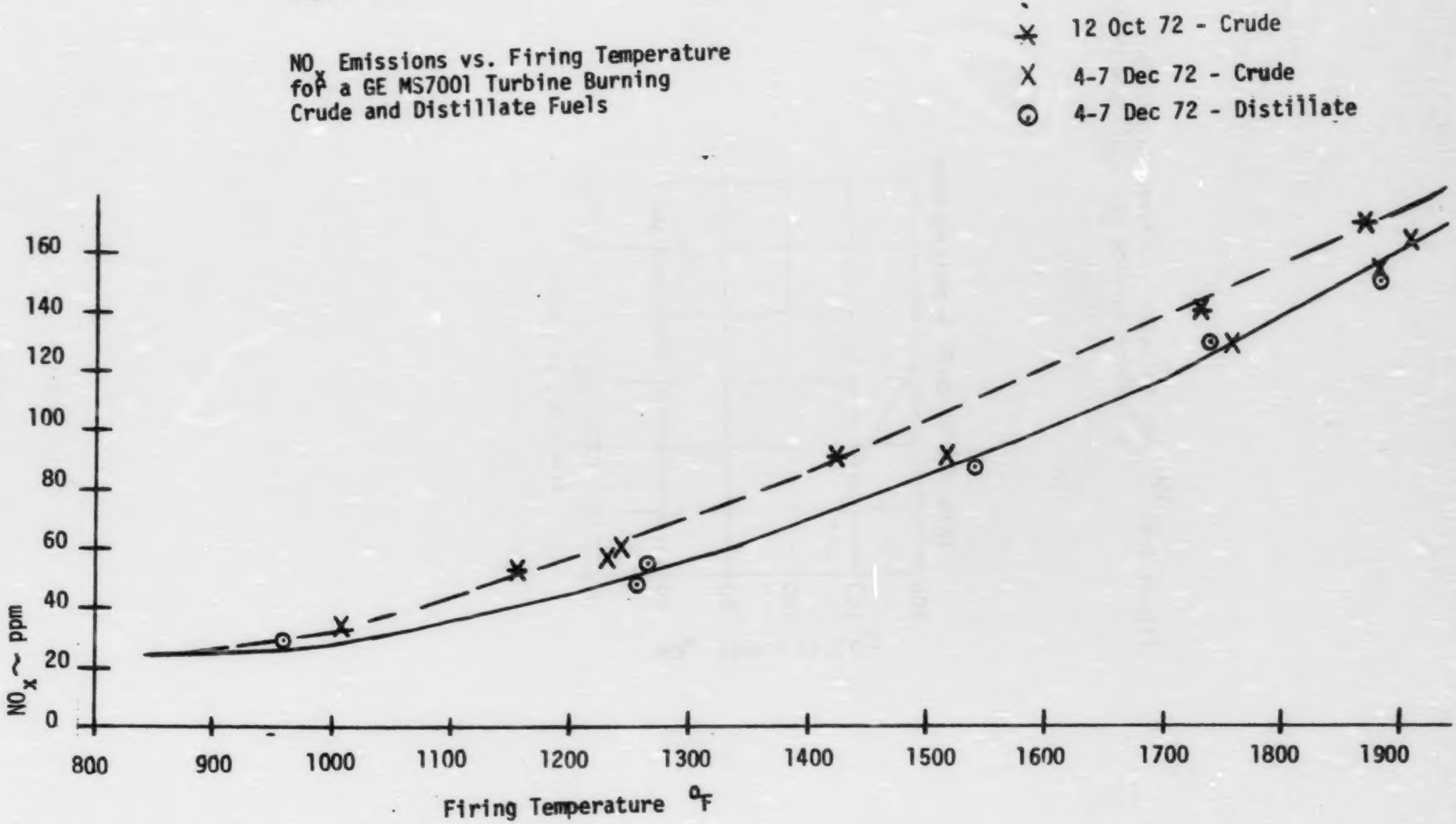
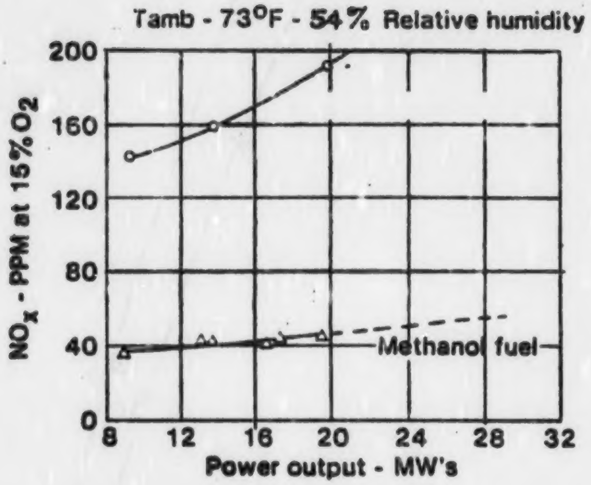


Figure 3-40 [156] NO_x Emission vs. Power Output for a Turbine Burning No. 2 Oil and Methanol



As noted in section 3.2.3.3.3, coal gas is an emerging fuel which has potential for use in gas turbine combustors. Figure 3-41 shows measured and predicted emissions of NO_x when burning natural gas, number 2 distillate and low Btu gases in a coal gas combustor. The agreement between predicted and measured values is close for the number 2 distillate oil and the 7.45 MJ/scm (200 Btu/scf) coal gas. The predicted NO_x emissions from burning the 4.84 MJ/scm (130 Btu/scf) coal gas are 3 ppm but this was not verified because measurement instruments could not resolve emissions below 10 ppm.[159] The curve showing the 11.18 MJ/scm (300 Btu/scf) gas is based on an oxygen blowing gasifier and, as can be seen from Figure 3-41, the NO_x emissions are predicted to be higher than when number 2 oil is burned. This predicted high level of emissions for the 11.18 MJ/scm (300 Btu/scf) gas is caused because the oxygen blown gasifier results in coal gases with high hydrogen and carbon monoxide contents which burn at higher flame temperatures than natural gas.[160] The reference therefore predicts that high levels of NO_x will result if pure oxygen is used as the oxidizing medium in a coal gasifier, but not if the medium is air.[161] Generally, the lower flame temperatures of low Btu fuels (e.g., air blown coal gas) will result in substantially reduced NO_x emissions as compared to natural gas or number 2 oil.[162, 163] This is also shown in Figure 3-42 for NO_x emissions from full and small scale combustors burning natural gas and coal gas. Lower NO_x emissions from low Btu gas as compared to number 2 distillate and natural gas have also been predicted based on laboratory combustor rig tests of a Texaco gasifier (100 - 150 Btu/scf gas) coupled with a Turbo Power and Marine FT4 combustor rig and the estimated emissions of NO_x for an FT4C-1 gas turbine based on results of these tests are shown in Figure 3-43.

Figure 3-41[158] Measured and Predicted NO_x Emissions from Burning Various Fuels in a Gas Turbine Coal Gas Combustor

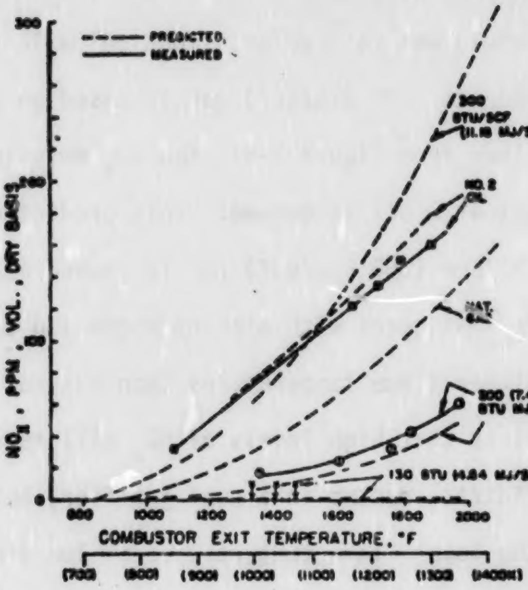
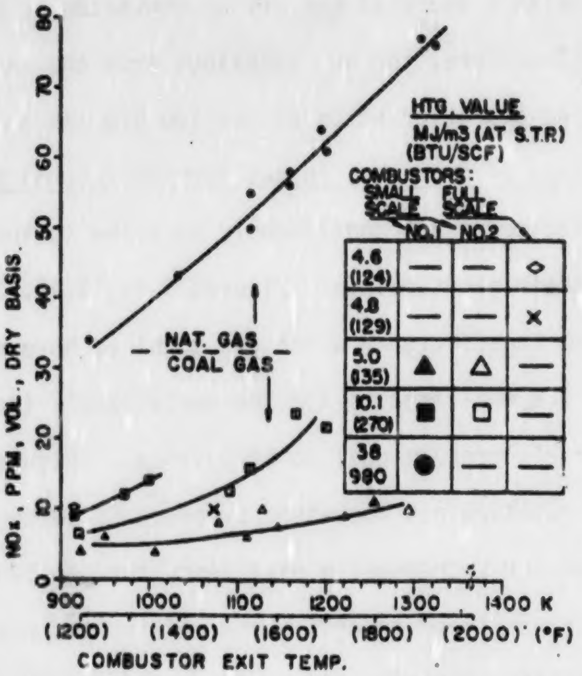


Figure 3-42[164] NO$_x$ Emissions from Full and Small Scale Combustors vs. Combustor exit temperature for Gas Turbine Combustors Burning Natural Gas and Coal Gas



As discussed in section 3.2.3.4, there are many design factors which affect the NO_x emissions from a gas turbine. These factors will also affect the NO_x emissions from a gas turbine burning low Btu gas. For example, the temperature of the coal gas used by the gas turbine will affect NO_x emissions by causing them to increase with increasing coal gas temperatures and any nitrogen compounds within the coal gas may be converted to NO_x as discussed in section 3.2.3.4.2. Therefore, the NO_x emissions from the gas turbine may be affected by the design characteristics of the low Btu gas generating system.

3.2.3.4.4 The effect of increases in gas turbine thermal efficiency on NO_x emissions - Manufacturers are continuously striving to increase the thermal efficiency of their gas turbines. Figures 3-44, 3-45, 3-46, 3-47, and 3-48 depict the increase in efficiency which gas turbines have experienced since 1960 and project the efficiencies for the early 1980's (heat rate as used in the figures is inversely proportional to efficiency). Figure 3-44 shows that the efficiency of gas turbines increased by about 40 percent from 1960 to 1975 and projects further increases in efficiency through 1980. Figures 3-46 and 3-47 illustrate the trend towards higher firing temperatures and compression ratios to obtain increased efficiencies from simple cycle gas turbines. Figure 3-48 illustrates the same trend toward higher compression ratios, firing temperatures and efficiency for unfired combined cycle gas turbines.

Figure 3-49 shows thermal efficiency versus specific output in Btu's of net output per pound of gas flow for simple cycle, regenerative cycle, and combined cycle gas turbines for firing temperatures of $1650^{o}F$ and $2000^{o}F$ and for different compressor pressure ratios. This figure shows that the efficiency of simple cycle machines and combined cycle machines increases with increases in pressure ratios and firing temperatures. The figure also

Figure 3-43[165] Estimated NO_x Emissions vs. Power Output For an FT 4C-1 Gas Turbine Burning Low Btu Gas, Natural Gas, and No. 2 Distillate

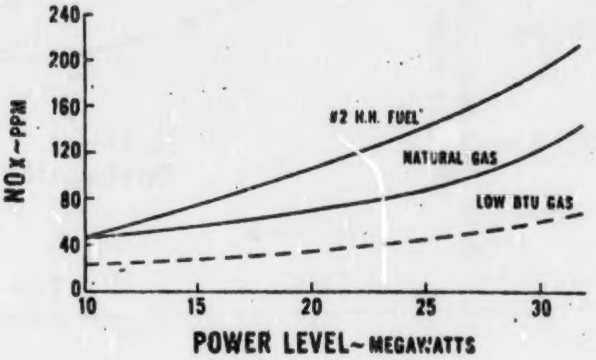
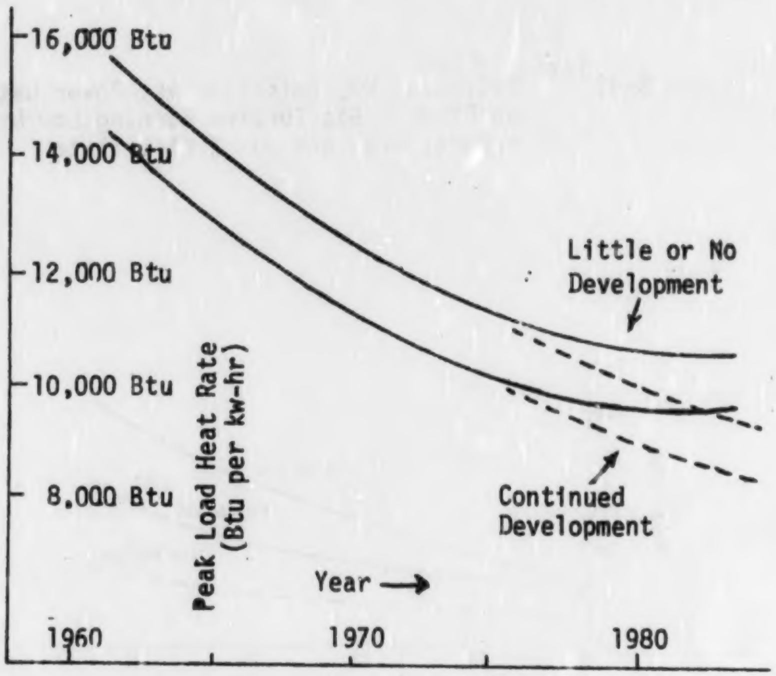


Figure 3-44[166] Improvement in the Heat Rate of Simple Cycle Gas Turbines Since 1960



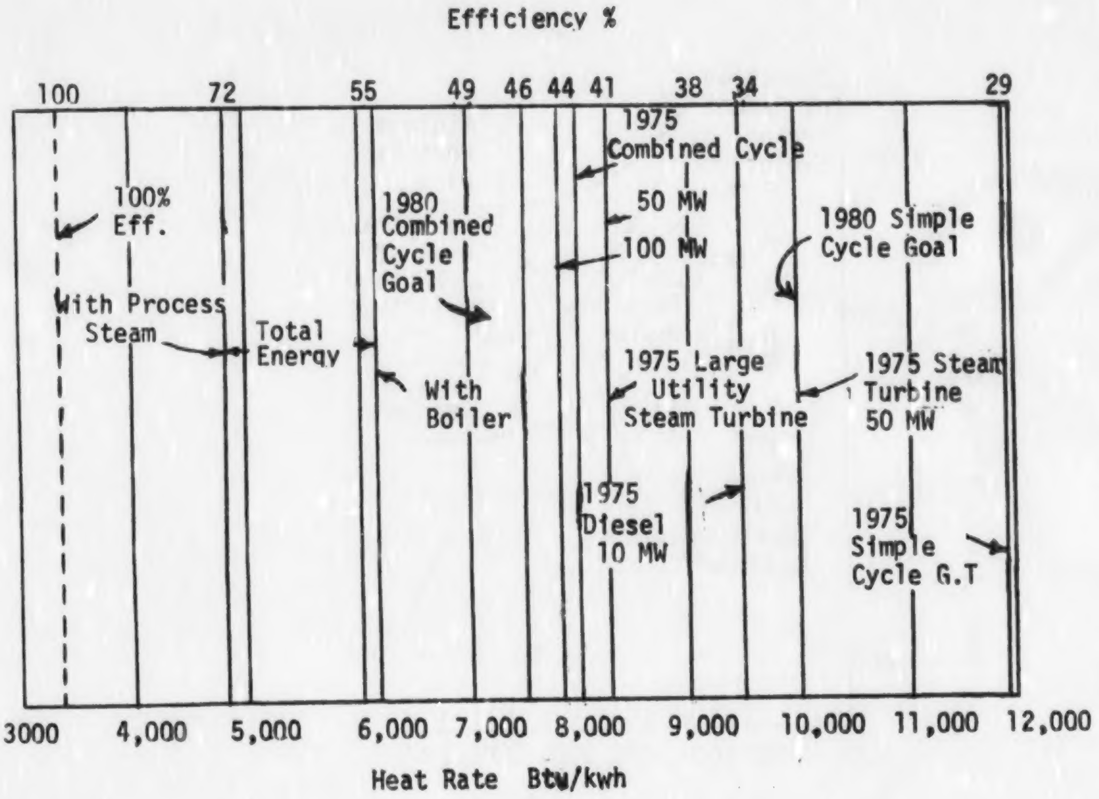


Figure 3-45[167] Simple Cycle and Combined Cycle Gas Turbine Heat Rates and Efficiencies for 1975 and Projections for 1980.

Figure 3-46[168] Projected HHV Heat Rates for Simple Cycle Gas Turbines vs. Firing Temperature for Various Compression Ratios

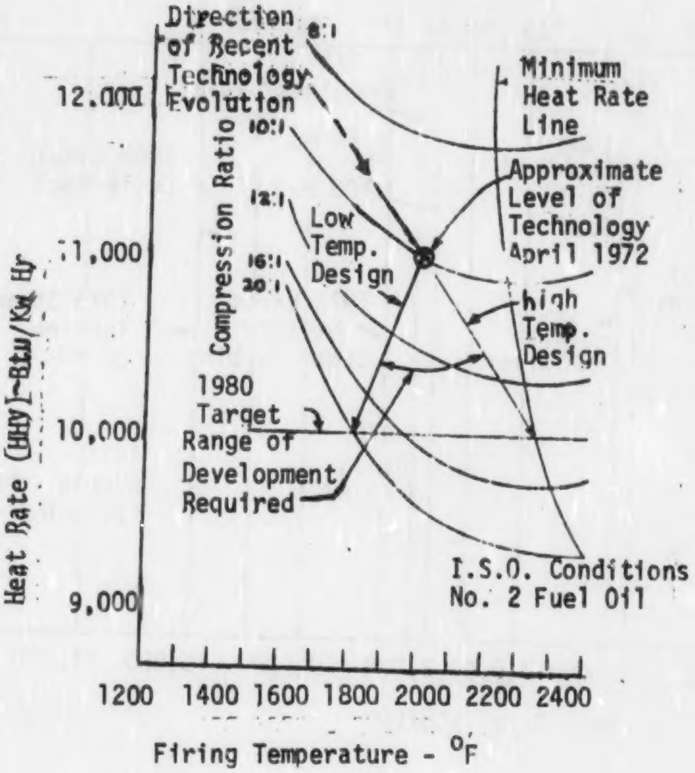


Figure 3-47[169] Heat Rate (HHV) vs. Time of Order for Simple Cycle Gas Turbines

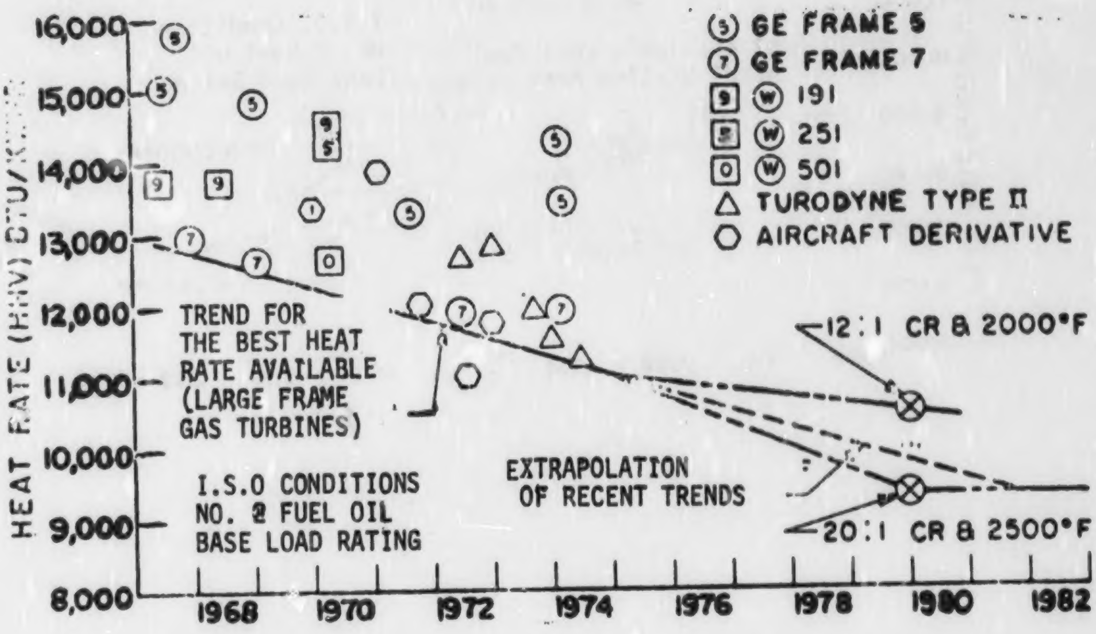


Figure 3-48[170] HHV Heat Rate vs. Time of Order for Unfired Combined Cycle Generating Units

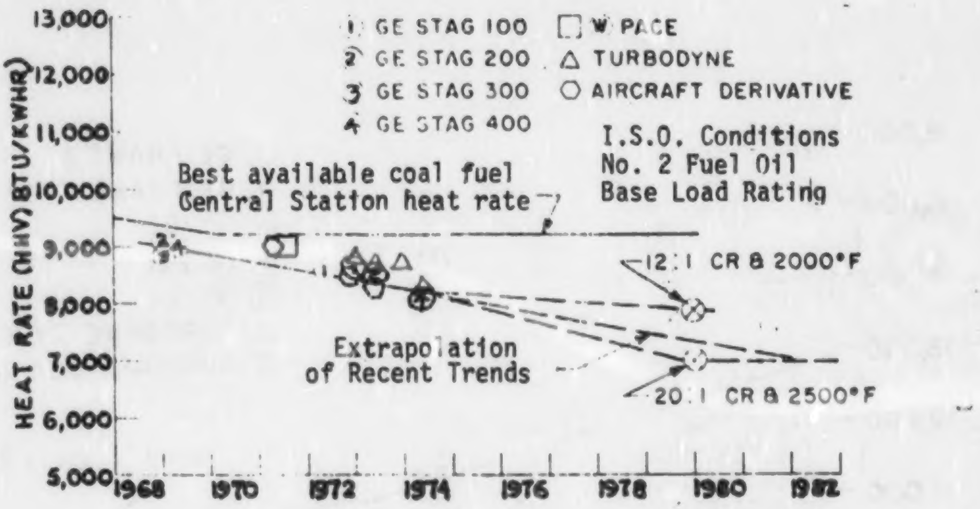
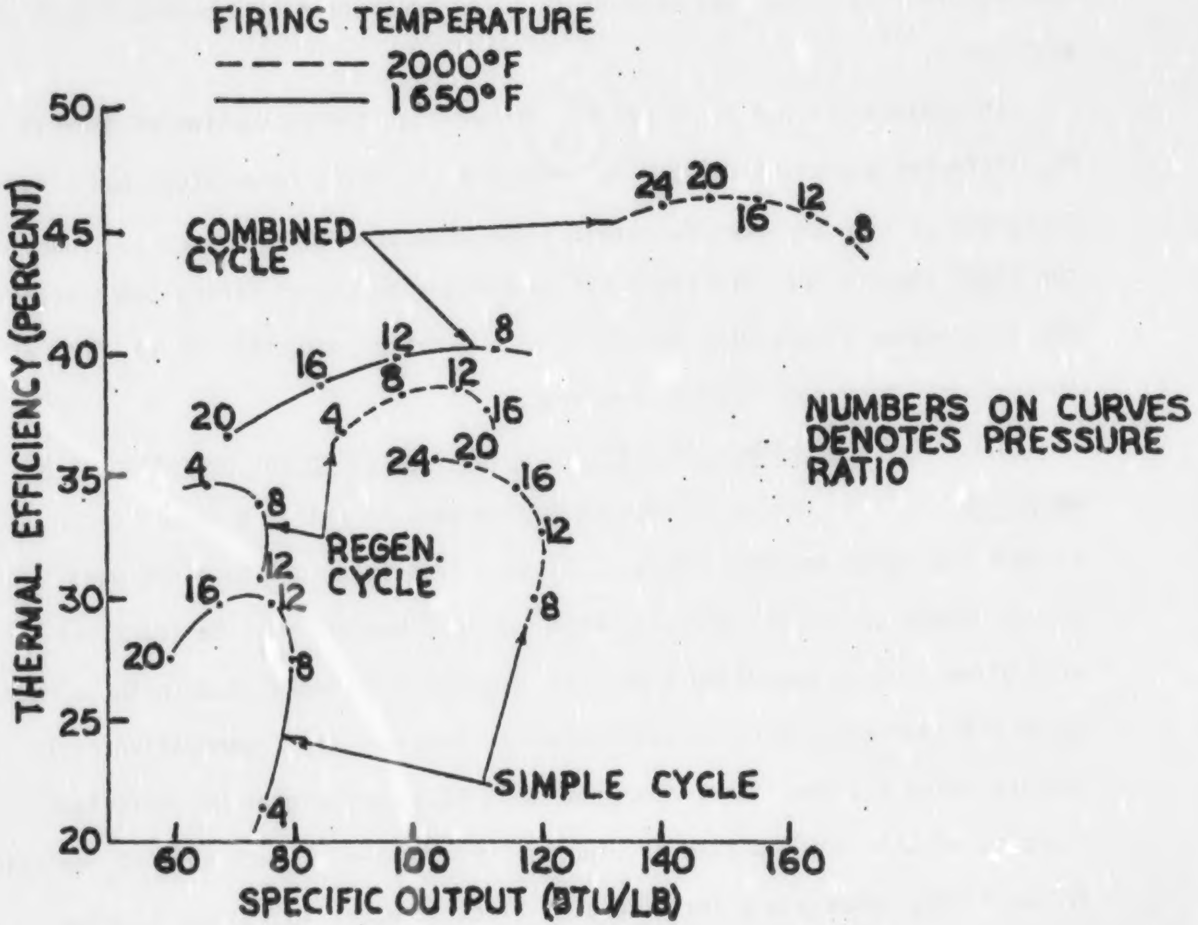


Figure 3-49[172] Gas Turbine Plant Performance for Combined, Regenerative and Simple Cycle Gas Turbines



shows the relationships between thermal efficiency, firing temperature, and compression ratio for regenerative machines and indicates that optimum efficiency for these machines occurs at higher firing temperatures with moderate compression ratios. This high efficiency at moderate compression ratios is due to the fact that the transfer of heat energy from the exhaust gases to the combustor inlet air (compressor outlet air) depends on the temperature differential between them and, therefore, a low compressor temperature rise (which corresponds to a low pressure ratio) provides optimum efficiency.[171]

In section 3.2.3.4.1, it has been shown that the production of thermal NO_x increases asympototically with increases in firing temperature and also increases as a square root function of the combustor inlet pressure. Therefore, the trend towards more efficient gas turbines with higher firing temperatures and compression ratios will result in much greater production of NO_x from a conventional type gas turbine combustor.

3.2.3.4.5 The effect of gas turbine configuration and operation on NO_x emissions - The same basic turbine can be used in either a simple cycle or combined cycle configuration. Emissions from identical turbines used in the simple cycle and combined cycle (unfired boiler) will be identical at a given turbine operating load. For identical turbines used in the simple cycle and regenerative cycle configuration, however, the regenerative cycle turbine using a conventional combustor will have much higher NO_x emissions. Examples of this are depicted in Figures 3-50 and 3-51, which show NO_x emissions versus firing temperature for a General Electric Model 7001 B gas turbine burning distillate oil and used in the simple cycle and regenerative cycle configurations, respectively. The much greater emission of NO_x from the regenerative cycle gas turbine is due to the increased combustor inlet temperature and the increased residence time of the combustion gases at high temperatures as described in section 3.2.3.4.1.

Figures 3-50 and 3-51 also depict the asympotic relationship between NO_x emissions and the firing temperature (which is directly related to combustion temperature) of the turbine. Since firing temperature is directly related to the power output of the gas turbine, Figures 3-50 and 3-51 show that NO_x emissions increase with increasing gas turbine load.

3.2.3.5 State and Local Regulations for Stationary Gas Turbines - Only a few state and local jurisdictions have regulations specifically for stationary gas turbines, although a considerable number have applied standards originally written for other sources to gas turbines.[175, 176] A summary of these regulations is shown in Table 3-10. Thus, the most stringent state regulation on NO_x from fuel combustion, which could be applicable to gas turbines of all sizes, is 0.2 lb NO_x/million Btu for gas fired burners and 0.3 lb NO_x/million Btu for oil fired burners. Eight states have this standard. San Diego County appears to have the most stringent NO_x regulations limiting emissions to 75 ppm (about 0.3 lb NO_x/million Btu) and 42 ppm at 15 percent oxygen when burning liquid and gaseous fuels, respectively.[178]

State sulfur regulations vary from 0.3 to 2.6 weight percent with an average of about 1 percent.[179] State visible emission regulations vary from 0 to 60 percent opacity with the typical standard, applicable to gas turbines in 40 states, restricting opacity to "less than 20 percent".[180] There are no known state regulations for CO emissions from gas turbines.

Table 3-11 is a tabulation of the typical and most stringent state or local standards to which gas turbines are subject. Table 3-12 shows the effect of such standards on emissions of NO_x and SO_2 from a typical large (33 megawatt) gas turbine.

Figure 3-50[173] NO$_x$ Emissions vs. Firing Temperature for an MS7001-B Simple Cycle Turbine Burning Distillate Fuel (I.S.O. Conditions)

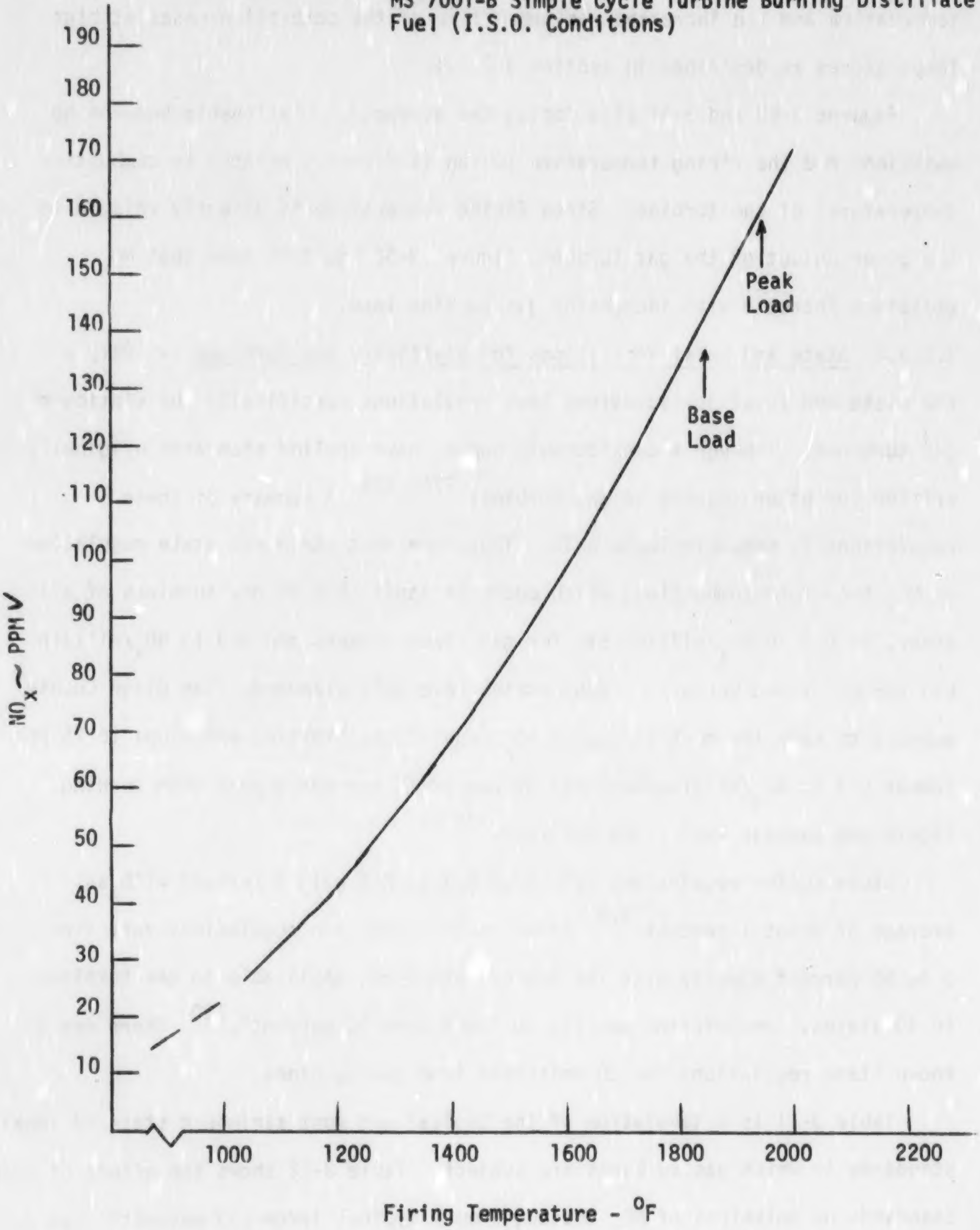


Figure 3-51 [174] NO_x Emissions vs. Firing Temperature for an MS 7001-B Regenerative Cycle Turbine Burning Distillate Fuel (I.S.O. Conditions)

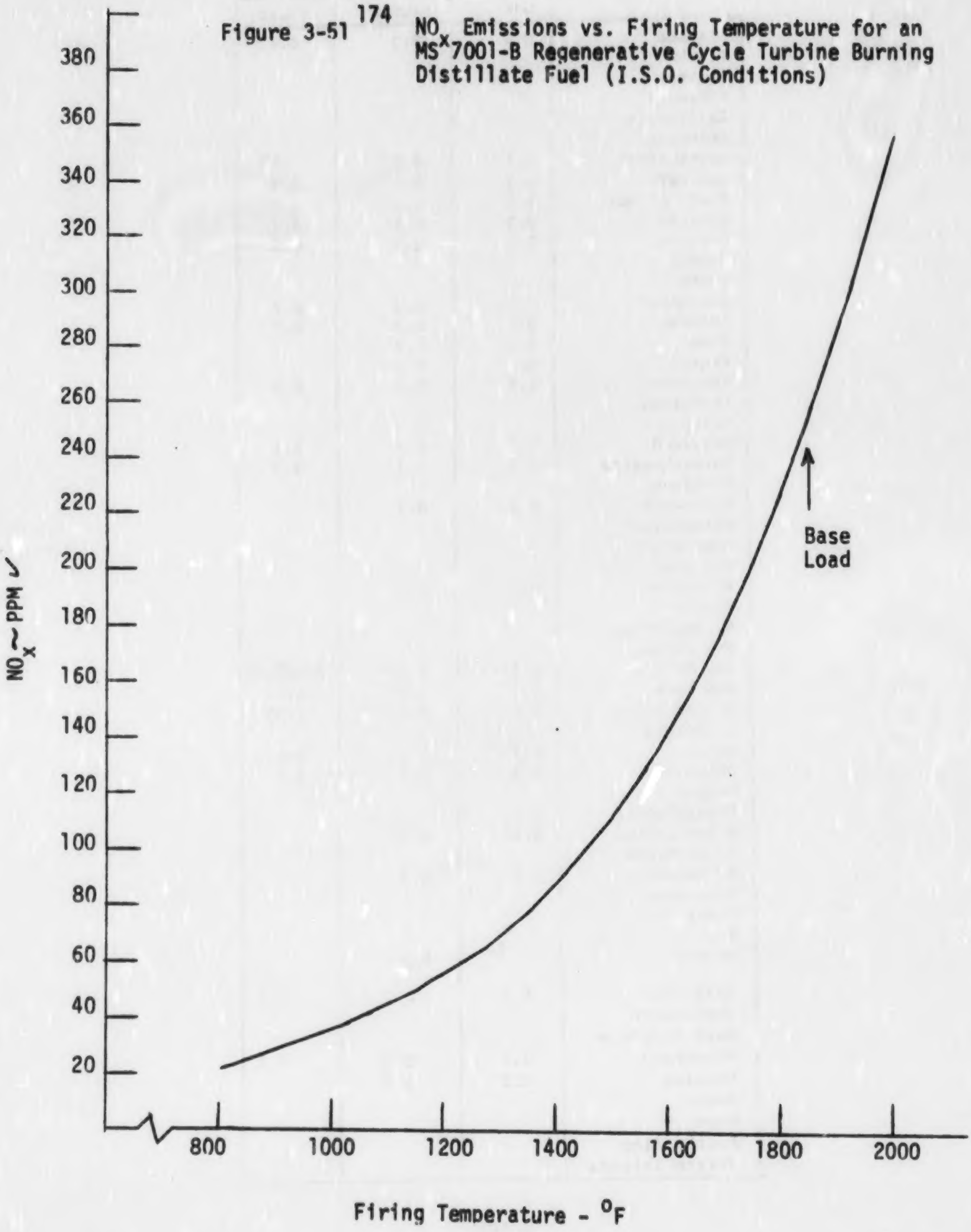


Table 3-10. Summary of State Regulations[177]

	FUEL BURNING LBS NO$_x$/10^6 BTU OF HEAT INPUT/HR		
	GAS	LIQUID	SOLID
Alabama	0.2	0.3	0.7
Alaska			
Arizona			
Arkansas			
California			
Colorado			
Connecticut	0.2	0.3	0.7[5]
Delaware	0.2	0.3	0.3
Dist. of Col.	0.2	0.3	0.7
Florida	0.2	0.3	0.7
Georgia	0.2	0.3	0.7
Hawaii			
Idaho			
Illinois	0.3	0.3	0.7
Indiana	0.2	0.3	0.7
Iowa	0.2	0.3	
Kansas	0.3	0.3	
Kentucky	0.2	0.3	0.7
Louisiana			
Maine			
Maryland	0.2	0.3	0.5
Massachusetts	0.3	0.3	0.3
Michigan			
Minnesota	0.2	0.3	
Mississippi			
Missouri			
Montana			
Nebraska	0.2	0.3	
Nevada			
N. Hampshire			
New Jersey			
New Mexico	0.2[13,14]	0.3[13]	0.45[2,15]
New York	0.2	0.3	0.7
N. Carolina	0.6	0.6	1.30
N. Dakota			
Ohio	0.2	0.3	0.9
Oklahoma	0.2	0.3	0.7
Oregon			
Pennsylvania	0.3	0.3	
Rhode Island	0.2[7]	0.3[7]	
S. Carolina			
S. Dakota	0.2	0.3	
Tennessee			
Texas			
Utah			
Vermont	[18]	0.3	
Virginia	0.2	0.3	0.7
Washington			
West Virginia			
Wisconsin	0.2	0.3	0.7
Wyoming	0.2	0.3	
Samoa			
Guam			
Puerto Rico			
Virgin Islands			

TABLE 3-11

STATE AND LOCAL EMISSION STANDARDS APPLICABLE TO GAS TURBINES

POLLUTANT	FUEL	TYPICAL	MOST STRINGENT
NO_x	GAS	0.3 LB NO_x/MBTU (75 PPM @ 15% O_2)	125 PPM @ 3% O_2 (42 PPM @ 15% O_2)
	OIL	0.3 LB NO_x/MBTU (75 PPM @ 15% O_2)	0.3 LB NO_x/MBTU (75 PPM @ 15% O_2)
SO_2	OIL	1% SULFUR BY WEIGHT (1 LB SO_2/MBTU)	0.3% SULFUR BY WEIGHT (0.3 LB SO_2/MBTU)
CO	ALL	NONE	NONE
VISIBLE EMISSIONS	ALL	20%	0%

TABLE 3-12

EFFECT OF EXISTING STATE AND LOCAL STANDARDS ON POLLUTANT EMISSION LEVELS FROM A 33 MW GAS TURBINE

		MAXIMUM EMISSIONS, LB/HR		
			CONTROLLED TO STATE/LOCAL STANDARDS	
POLLUTANT	FUEL	NO CONTROLS	TYPICAL	MOST STRINGENT
NO_x	GAS	170 ppm 251	124	66
	(distillate) LIQUID	230 ppm 373	124	124
SO_2	LIQUID	2% S 826	413	124

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4. EMISSION CONTROL TECHNIQUES

4.1 PARTICULATE EMISSIONS

4.1.1 General

As discussed in sections 3.2.2.1 and 3.2.2.2, particulate emissions from gas turbines consist of ash from the fuel and particulates of carbon and hydrocarbons resulting from incomplete combustion. Fuels containing high ash and vanadium contents will result in higher particulate emission rates than light distillate fuels or natural gas. As discussed in section 3.2.2.1, particulate emissions from a turbine operating at 20 MW and burning natural gas were about 0.2 lb/MWH compared to 0.8 lb/MWH for a turbine operating at 52 MW and burning #2 oil. Because of the high gas velocities, extreme turbulence caused by the ducting configuration and sound baffles, and the difficulty of sampling with very long probes in the environment of the exhaust gases (as discussed in section 3.2.2.1), measurement of particulate emissions from gas turbines are of questionable accuracy and high variability. Magnesium, manganese and barium inhibitors added to liquid fuels to retard vanadium corrosion of the turbine components will also increase particulate emissions.[1,2,3]

4.1.2 Control Techniques for Particulate Emissions from Gas Turbines

Particulates are emitted from gas turbines at low levels; varying from 0.002 gr/scf to 0.10 gr/scf and exhaust gas volumes ranging from 166,206 scfm to 460.167 scfm for turbines operating at base loads of 12 and 44 megawatts, respectively.[4,5,6,7]

As discussed in sections 3.2.2.1, 3.2.2.2, and 3.2.2.3, particulate emissions may be decreased by burning natural gas or low ash fuels and by combustor modifications which provide more complete combustion of hydrocarbons and carbonaceous particles.

The only known application of a particulate control device for gas turbines has been the installation of a packed bed scrubber on several jet-engine test cells at a naval air station. This scrubber reportedly reduced particulate emissions from uncontrolled levels ranging from about 0.008 gr/scf to 0.015 gr/scf, depending on engine operating mode, to controlled levels ranging from 0.001 gr/scf to 0.005 gr/scf.[8,9] Opacity after the scrubber was generally less than 10%.

A jet engine test cell is not typical of a stationary gas turbine installation. Because a jet engine operates in different modes, including an afterburner mode, opacities vary from 0 to 80 percent whereas a typical stationary gas turbine will operate with opacities generally less than 20 percent. The scrubber discussed above was installed to reduce the opacity, noise and particulate emissions to meet requirements for Federal facilities. A brief discussion of the costs which would be incurred to install such a scrubber on stationary gas turbines is presented in Chapter 7.

4.2 Visible Emissions

The factors which affect visible emissions have been discussed in section 3.2.2.2.1. Visible emissions are related to particulate emissions so the factors discussed in section 4.1 to reduce particulate emissions from stationary gas turbines may also reduce visible emissions.

One way to reduce visible emissions is to burn fuels with high hydrogen contents. This is shown by the data in figures 3.13 and 3.14. Another method for reducing visible emissions is to use fuel additives such as soluble compounds of manganese, barium, lead, iron and others. The actual mechanism by which these additives reduce visible emissions is unknown and somewhat controversial. For example, a paper by personnel of Ethyl Corporation states that the mass of particulate emissions is reduced by the use of manganese additives while a paper by personnel of General Electric does not support such a conclusion.[10,11]

Major reductions in visible emissions have been achieved through combustor redesign. The design of low visible emission combustors involves providing leaner fuel-to-air mixtures and more affective fuel and air mixing in the primary zone.[12,13,14] Figure 4.1 shows visible emission data for General Electric aircraft turbines developed both prior to 1966 and after 1966. The figure shows that visible emissions from engines developed after 1966, which incorporate combustor design modifications to reduce visible emissions, are below the threshold of visibility. The newer models of stationary gas turbines have also incorporated combustor modifications such that visible emissions are below the visible threshold and/or are below 10 percent opacity for turbines burning fuels ranging from distillates to residuals.[16,17,18,19] Figure 4.2 shows the relative effect of using a fuel additive versus combustor modifications to reduce visible emissions. Visible emissions increase exponentially with Bacharach number (e.g. to reduce visible emissions from Bacharach 9 to 4 requires eight times the smoke reduction as that to reduce visible emissions from Bacharach 5 to 4.)[20] Figure 4.2 shows that combustor modifications reduced emissions substantially while the use of the manganese additive had relatively little effect.

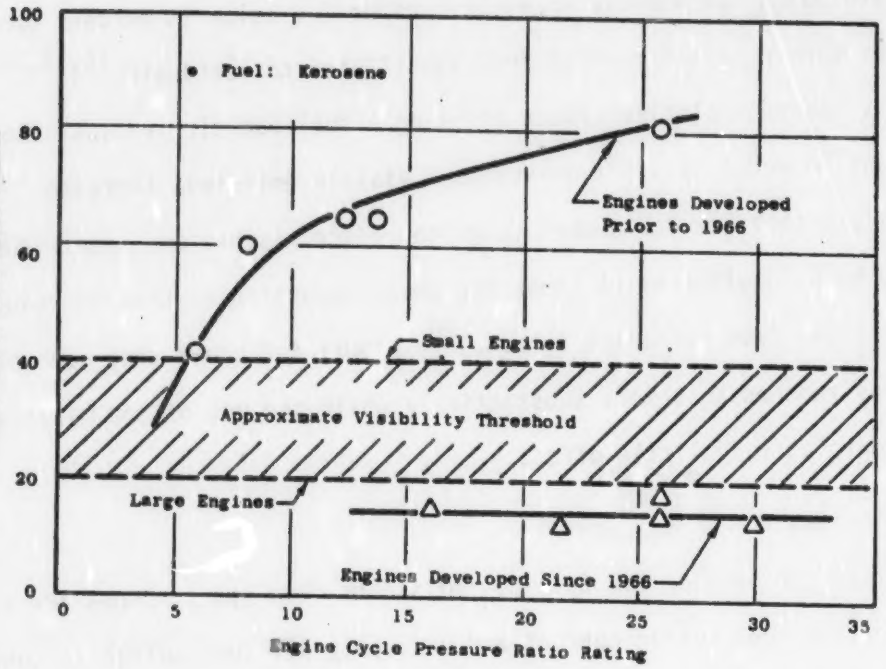
4.3 SO_2 Emissions

As stated in section 3.2.2.1, SO_2 emissions from gas turbines are strictly a function of the fuel sulfur content and virtually all fuel sulfur is converted to SO_2. The only technique now used to control SO_2 emissions from gas turbines is to burn low sulfur fuels. Stack gas scrubbing for SO_2 removal has not been applied to gas turbines primarily because it is less costly to desulfurize the fuel. Relative costs are briefly presented in Chapter 7. Figure 4.3 shows emission rates from gas turbines for various fuel sulfur contents and for varying fuel flow rates.

Distillate fuels range in sulfur content from 0.01 to 0.48 percent by

Figure 4.1[15]

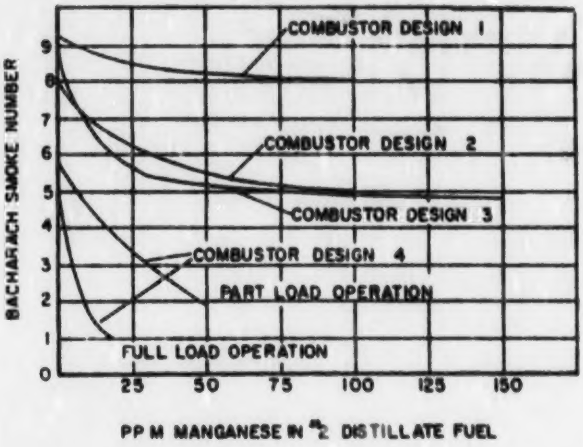
Comparison Of Peak Engine Smoke Emission Characteristics Of Various General Electric Aircraft Turbine Engines, At Standard Day-Sea Level Static Operating Conditions



Note: Low smoke numbers indicate low visible emission levels.

Figure 4.2[21]

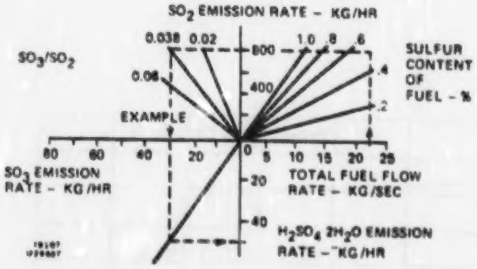
The Effect of Fuel Additives and Combustor Redesign on Visible Emissions



Note: Bacharach smoke number decreases with decreasing visible emissions.

Figure 4.3[21]

SO_2 Emission Rate as a Function of Fuel Sulfur Content and Gas Turbine Fuel Flowrate



weight; crudes contain from 0.06 to 3.0 percent by weight; and residual oils vary from 0.5 to 3.2 percent by weight.[22,23] Gas turbines have historically burned the low sulfur distillate fuels or natural gas. During the early 1970's there was a trend towards the installation of fuel treatment facilities to treat residual and crude oils so they could be fired in gas turbines. These treatment facilities removed the salts from the fuels and added inhibitors to prevent the vanadium in the fuel from corroding the gas turbine components but did not reduce the sulfur content. This trend toward the use of residual and crude oils is not evident today and the use of treatment facilities already installed has been minimal.[24,25,26,27] In a September 1975 survey of electric utilities, only four of 48 respondees indicated plans to burn residual oil in their gas turbines.[28] In response to a questionnaire distributed to electric utilities by Edison Electric Institute in October 1975, 17 of 17 respondees indicated that no trend existed to use crude or residual oils for firing gas turbines.[29] Therefore, it appears that the low sulfur distillate oils will continue to be burned by the owners and operators of gas turbines as a means of controlling SO_2 emissions and that high sulfur crude oils and residual oils will not be used. The primary reason for the shift away from crude and residual fuels is one of economics. In todays market, it simply costs more to buy and treat the crude and residual oils than to purchase and burn the distillate oils.[30]

In the event that the trend to crude and residual oils again emerges, the control technique for reducing SO_2 emissions is to remove the sulfur from the oil prior to its combustion in the turbine.

4.4 HYDROCARBON AND CARBON MONOXIDE EMISSIONS

4.4.1 General

As shown in figures 3.17 and 3.18, hydrocarbon and carbon monoxide emissions from conventional gas turbines are low at full load operation and increase asymptotically at low turbine loads. In section 3.2.2.3.1, it has been shown that incomplete combustion is the principal cause of hydrocarbon and carbon monoxide emissions from gas turbines. Most existing turbines are designed for maximum efficiency at full load operation with efficiency dropping to 90-95 percent at low load operation as discussed in 3.2.2.3.1.

Gas turbines with higher combustion efficiencies tend to have lower CO emissions as can be observed by comparing the heat rates (inverse of efficiency) and CO emissions for the turbines shown in Table 3.5. The CO emissions versus turbine size for small and large gas turbines without NO_x controls are shown in Figures 4.4 and 4.5, respectively. The data show that no generalization can be made regarding CO emissions versus turbine size. It is generally accepted as a "rule of thumb" that, for large gas turbines, a 1 percent decrease in turbine efficiency is incurred for each 300 ppm of CO emissions.[31] HC emissions also increase with decreasing combustion efficiency. Figures 4.6 and 4.7 show HC emissions versus turbine size for small and large turbines, respectively. These figures show that most gas turbines, regardless of size, have HC emissions generally less than 20 ppm when operated at or near full load. As explained in section 3.2.2.3.1, combustion chemical kinetics data show that vaporized hydrocarbons and any partially oxidized hydrocarbons are consumed during combustion much more rapidly than CO and thus the major product of inefficiency is expected to be CO.[32] This is demonstrated by data from an Airesearch GTC 85-90 gas turbine used in ground cart applications.[33]

Turbine load	horsepower	0	121	168	192
Combustion efficiency	%	97.2	98.6	99.3	99.6
Hydrocarbons as CH_4	lbs/hr	1.576	0.627	0.141	0.061
HC	ppm @ 15% O_2	495	134	26	10
Carbon Monoxide	lbs/hr	7.37	7.21	5.38	3.34
CO	ppm @ 15% O_2	1324	880	566	315

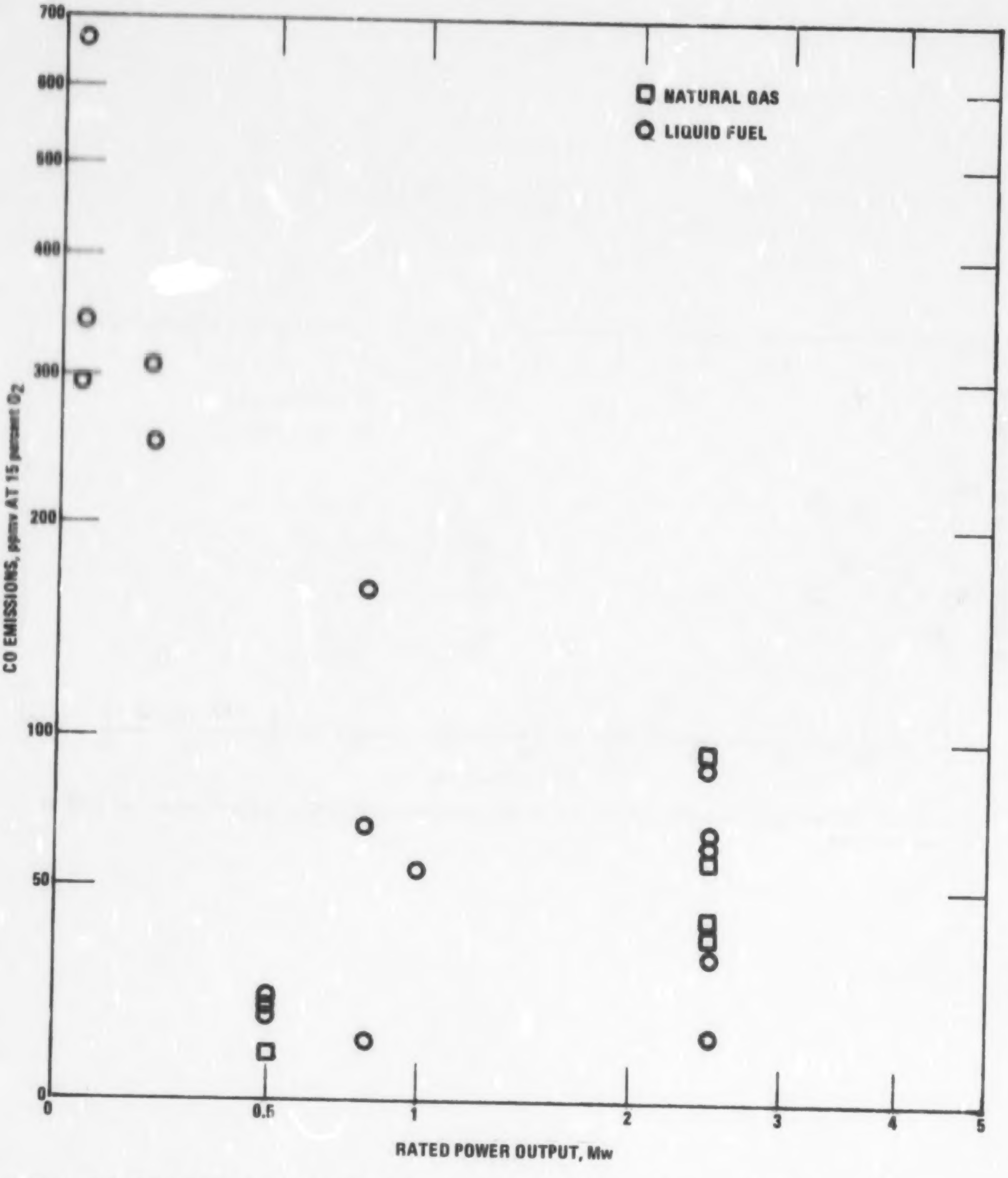


Figure 4.4. CO emissions vs turbine size for small gas turbines without NO_x controls when operated at or near full load.

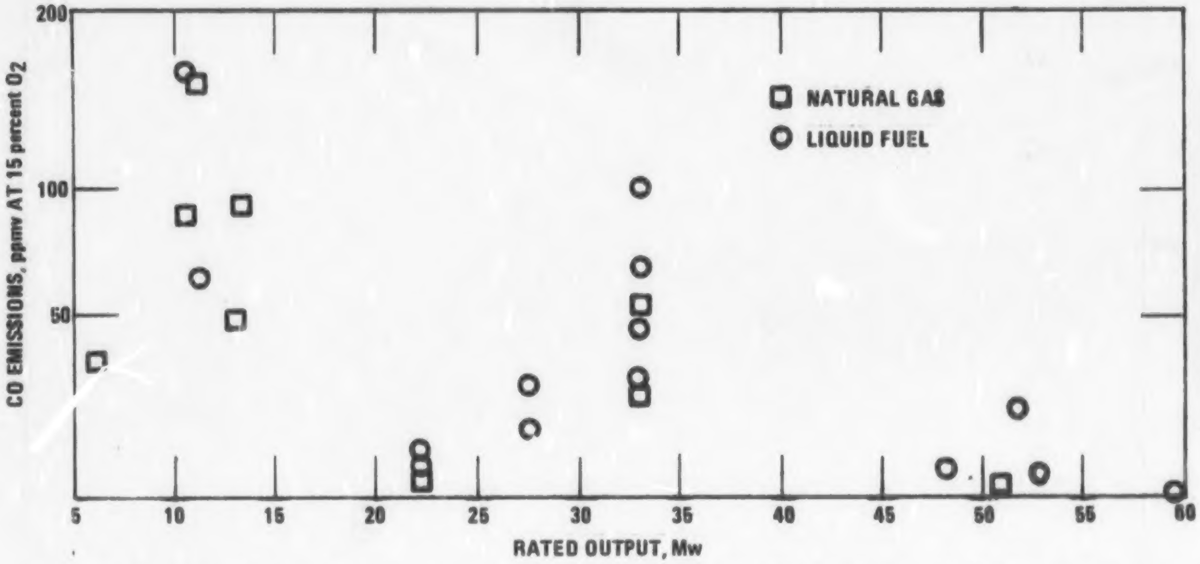


Figure 4.5. CO emissions vs turbine size for large gas turbines without NO_x controls when operated at or near full load.

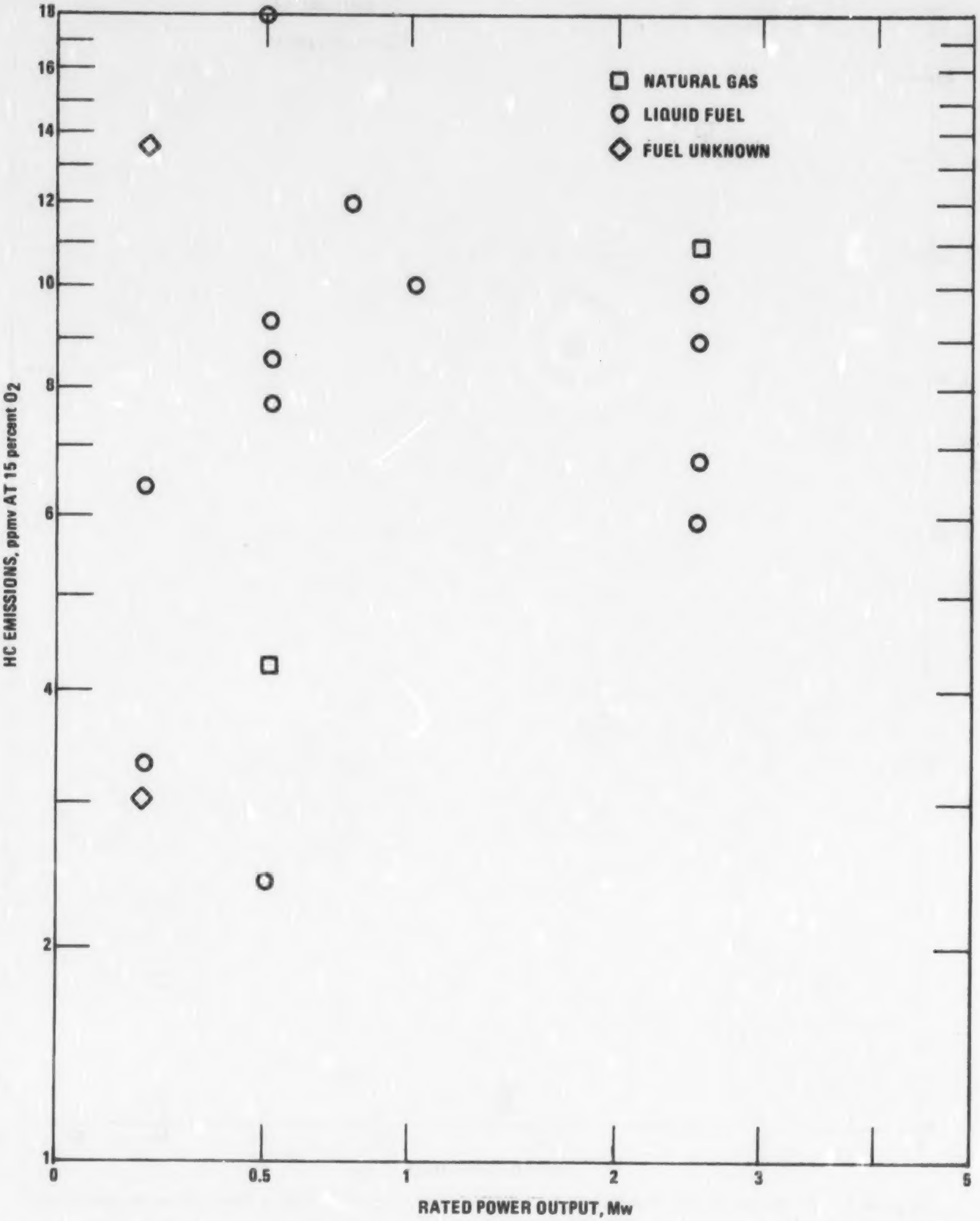


Figure 4.6. HC emissions vs turbine size for small gas turbines without NO_x controls when operated at or near full load.

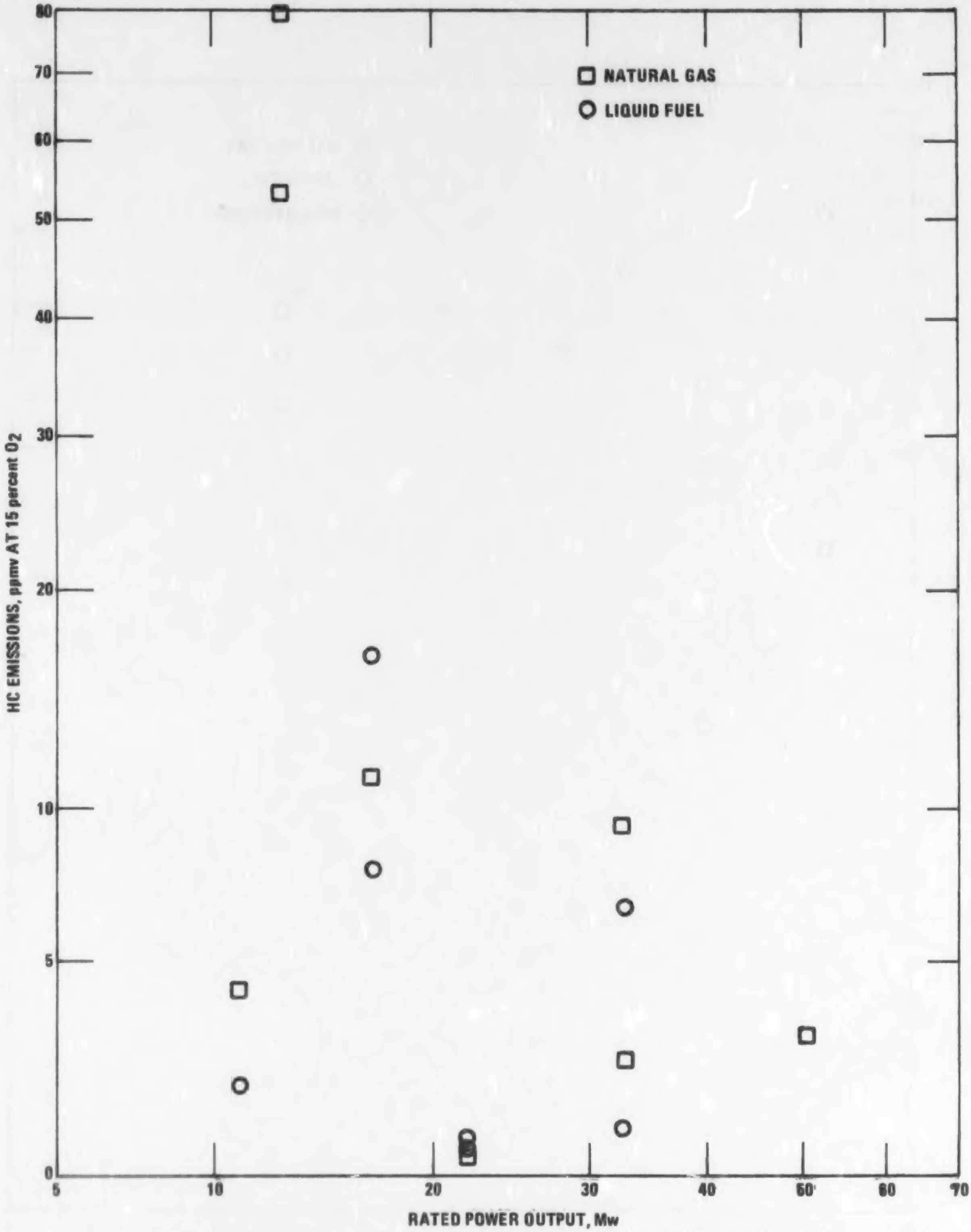


Figure 4.7. HC emissions vs turbine size for large gas turbines without NO_X controls when operated at or near full load.

These data again illustrate the trend for CO and HC emissions to decrease with increasing load and show that HC emissions are generally much lower than CO emissions.

4.4.2 Control of HC and CO Emissions

Since combustion efficiencies are high for turbines operating at base load conditions, HC and CO emission levels from stationary gas turbines are usually very low as noted in Chapter 3. As explained in section 3.2.2.3.1, the control of HC and CO emissions from stationary gas turbines, therefore, normally becomes important only at low loads where combustion efficiency decreases for gas turbines using conventional combustors. At low loads; low combustion inlet air temperatures cause quenching, low fuel to air ratios result in low equivalence ratios and reduced burning intensity in the primary zone, and the low fuel and air flows result in poor fuel atomization and distribution.

The control of HC and CO emissions is primarily a function of fuel injection and atomization and fuel-air mixing. Decreased HC and CO emissions are therefore accomplished by combustor and fuel injection modifications which promote better fuel atomization, better fuel and air mixing, and by controlling the fuel to air ratios and residence time at temperature, as necessary to provide combustion of the HC's in the primary zone of the combustor and combustion of the CO in the primary and intermediate zones of the combustor.

Unlike their stationary counterparts, gas turbines used in aircraft are commonly operated at low power levels while the aircraft is on the ground. Therefore, high levels of CO and HC are emitted at airports during these idle conditions and the more recent efforts to control HC and CO have been targeted at idle conditions. These data are discussed to illustrate the reductions which are achievable through application of combustor modifications.

Figure 4.7 presents CO and HC emission data reported by General Electric from tests of combustors for aircraft turbines. The figure demonstrates about a 60 percent reduction in HC emissions and a 40 percent reduction in CO emissions at idle power, when improved fuel atomization techniques are applied. Figure 4.8 shows that the CO and HC emissions are reduced by 75 and 90 percent, respectively, when fuel staging (sector burning) techniques are applied at idle power to an annular combustor used in aircraft turbines. The figures shown here and in the remainder of this section on HC and CO emissions present these emissions as grams/kilograms of fuel.1/ 2/

Another method of reducing CO and HC emissions at low turbine loads without adversely affecting turbine performance at high power conditions is to bleed off a portion of the compressor discharge air before the combustor. This controls the fuel-to-air ratio in the combustor. Figure 4.9 demonstrates that CO and HC emissions from an annular combustor operating at idle conditions were reduced by about 25 and 55 percent, respectively, when 20 percent of the compressor discharge was bled overboard.

An "Experimental Clean Combustor Program" funded by the National Aeronautics Space Administration and performed separately by General Electric and Pratt & Whitney has a primary objective to identify, define and develop low emissions combustors for advanced aircraft engines. A more complete description of the combustor concepts tested is contained in section 4.5 - NO_x Emissions. Phase I of the effort by General Electric identified several R & D combustor configurations which resulted in reduced HC and CO emissions. Fuel staging at idle was a significant factor in all the design approaches. Data for HC and CO emissions at idle conditions are presented in Figure 4.10 for the major design approaches used in the investigation. In Phase I, GE identified

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1/ ppm of CO at 15% O_2 ≈ 20 (Emission Index)

2/ ppm of HC as CH_6 at 15% O_2 ≈ 32 (Emission Index)

Figure 4.7[34]

CO and HC Emission Reductions with Improved Fuel Atomization in an Aircraft Gas Turbine at Ground Idle Power Operating Conditions

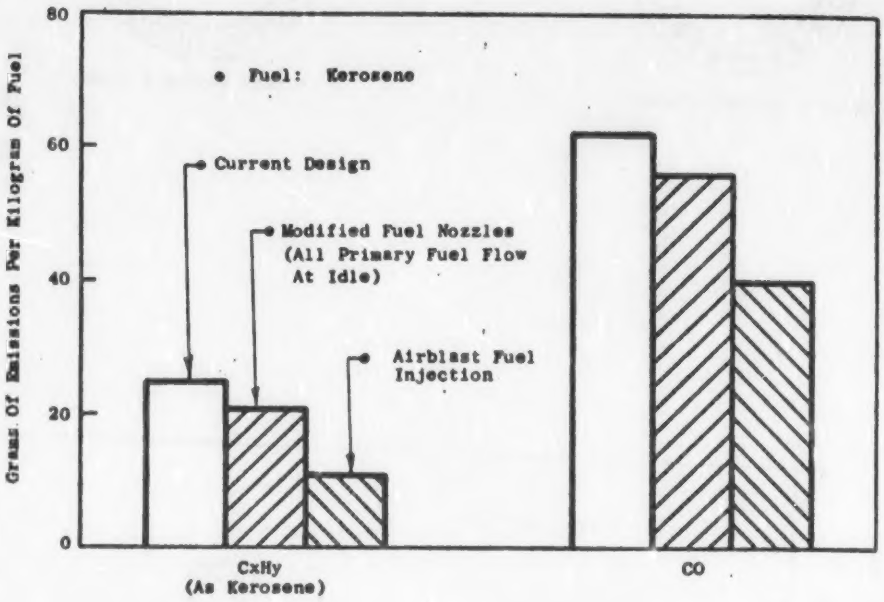
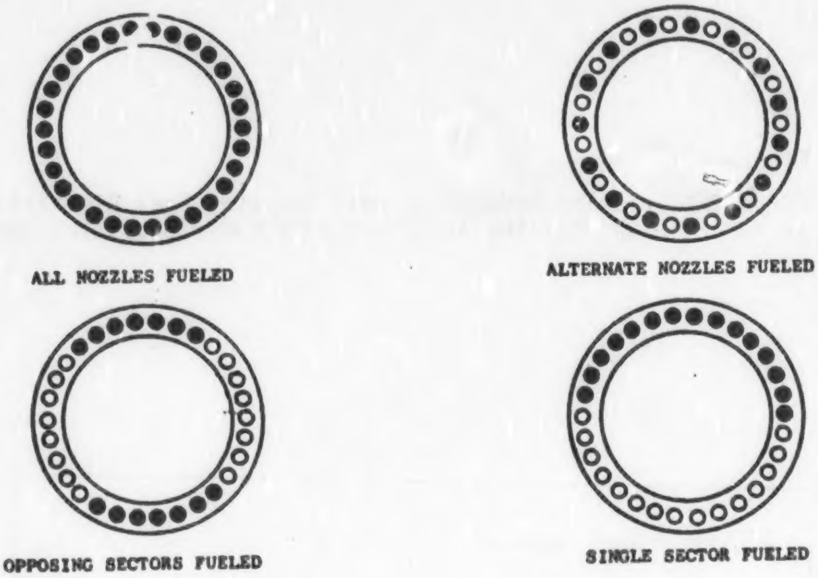


Figure 4.8[35]

CO and HC Emissions Level Reductions in an Aircraft Gas Turbine at Ground Idle Power Conditions Using Various Concepts of Staged Fuel Injection


ALL NOZZLES FUELED
ALTERNATE NOZZLES FUELED
OPPOSING SECTORS FUELED
SINGLE SECTOR FUELED

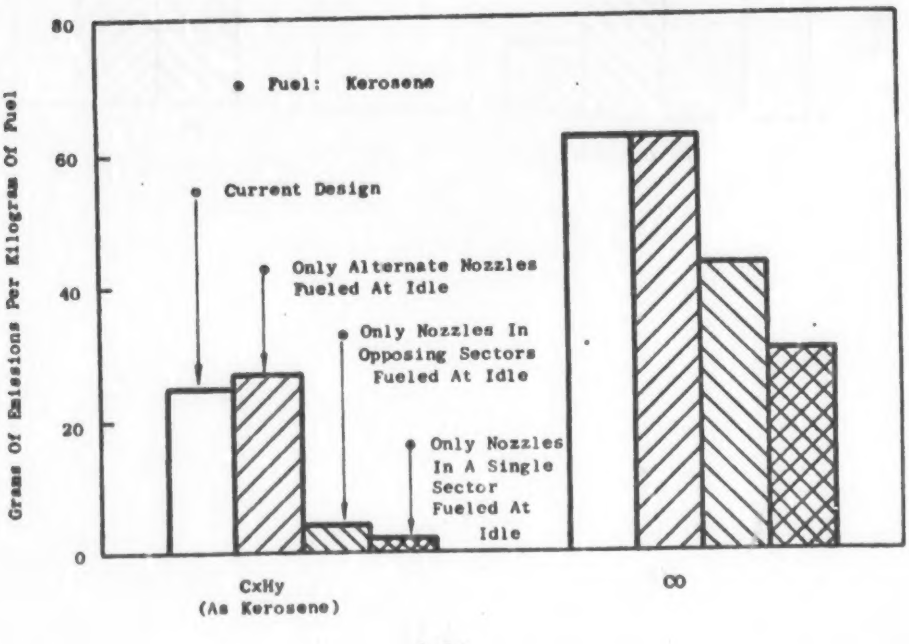


Figure 4.9[36]

CO and HC Emissions Level Reductions in an Aircraft Gas Turbine, at Ground Idle Power Operating Conditions, Standard Day-Sea Level Static vs. Compressor Bleed Air Extraction

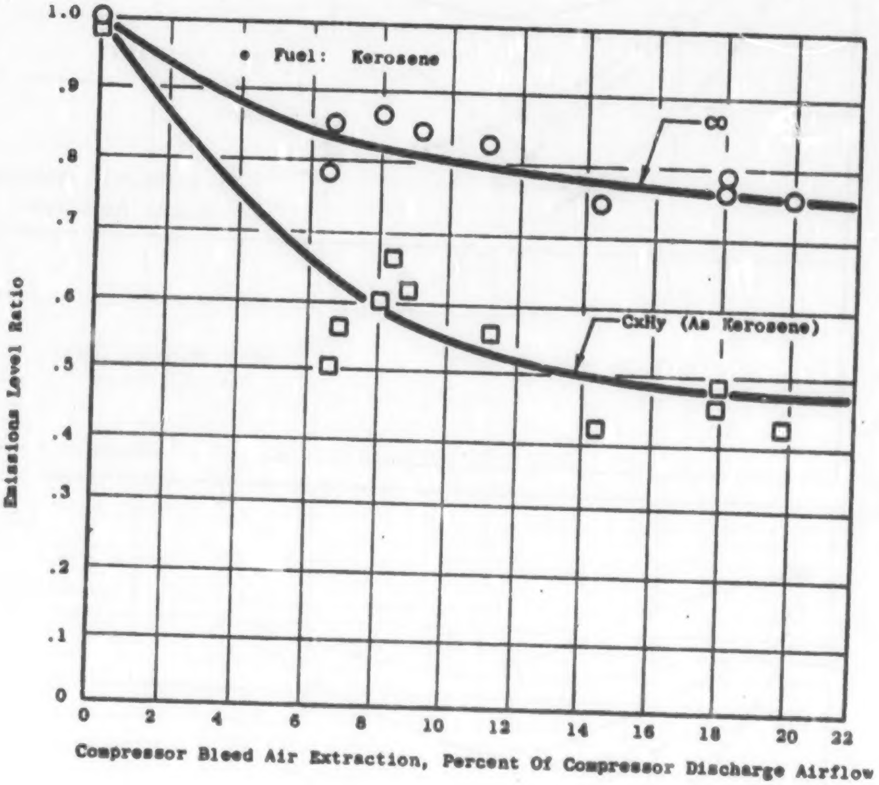
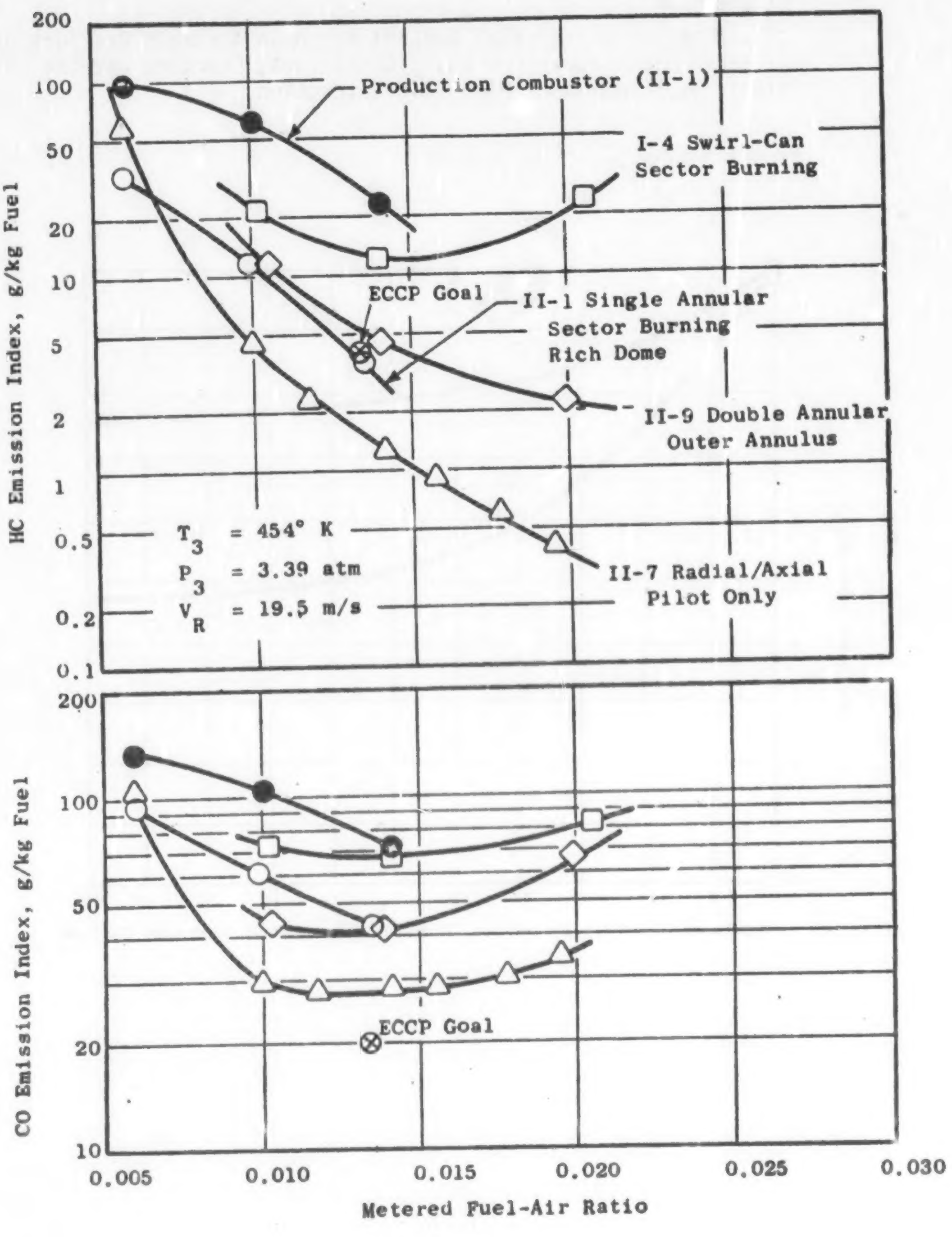


Figure 4.10[37]

CO and HC Emissions Reduction Using Various Combustor Design Modifications (combustor rig tests)



the double annular combustor and the radial/axial staged combustor as the two most promising configurations for reduction of NO_x, CO and HC. For the combustor rig tests (data shown in figure 4.10), the radial/axial staged combustor exhibited CO and HC reductions of about 58 and 93 percent, respectively, over the current production combustor. The double annular combustor exhibited CO and HC reductions of about 49 and 78 percent, respectively, over the current production combustor.

Data for CO and HC emissions at idle conditions from the best configurations of three design approaches investigated by Pratt & Whitney are shown in Table 4.1. This table shows that the swirl can configuration did not result in reduced emissions as compared to the JT9D-7 production combustor. The staged premix configuration P-3 reduced CO and HC idle emissions by 88 and 97 percent, respectively, as compared to the JT9D-7 production combustor. The swirl Vorbix configuration S3 did not significantly reduce CO or HC emissions, but configuration S8 reduced CO and HC emissions by about 61 and 85 percent, respectively, as compared to the JT9D-7 combustor.

It must be stressed again that these combustors for which HC and CO emission reductions have been discussed are designs to reduce the HC and CO emissions at low loads for turbines which already have extremely low HC and CO emissions at full load as demonstrated in Figures 4.11 and 4.12. The data illustrate, however, the viability of the control techniques.

4.4.3 Summary of Control Techniques for HC and CO Emissions

The data presented in this section show that combustion modifications have been demonstrated which reduce CO emissions at idle power by 25 to 75 percent and HC emissions at idle power by 55 to 93 percent. Turbines operating at load have been shown to have minimal CO and HC emissions.

Table 4.1[38]

SUMMARY OF CO & HC EMISSIONS FOR BEST CONFIGURATIONS OF EACH COMBUSTOR CONCEPT AT IDLE CONDITIONS

Combustor Concept and Configuration	Operating Condition	Emission Index (g/kg fuel) Carbon Monoxide	Emission Index (g/kg fuel) Total Unburned Hydrocarbons	Combustion Efficiency (Percent)
Goals	With and without bleed	20	4	99.1
Production JT9D-7 Combustor	Without bleed	77	29.8	94.7
Swirl Can Configuration N3 (Inner row of carburetor modules only)	With bleed	117	61	90.1
Swirl Can Configuration N11 (Outer row of carburetor modules only)	Without bleed	78.5	58.5	91.3
Staged Premix Configuration P3	With bleed	9	1	99.8
Staged Premix Configurations P7,P8	Without bleed	Unstable at design point		
Swirl Vorbix Configuration S3	With bleed	68	29	95.0
Swirl Vorbix Configuration S8	Without bleed	29	4.5	98.8

Notes: Combustor rig conditions with bleed were inlet pressure of 2.93 atm, inlet temperature of 428 K, and fuel-air ratio of 0.0126.

Combustor rig conditions without bleed were inlet pressure of 3.74 atm, inlet temperature of 456 K, and fuel-air ratio of 0.0105.

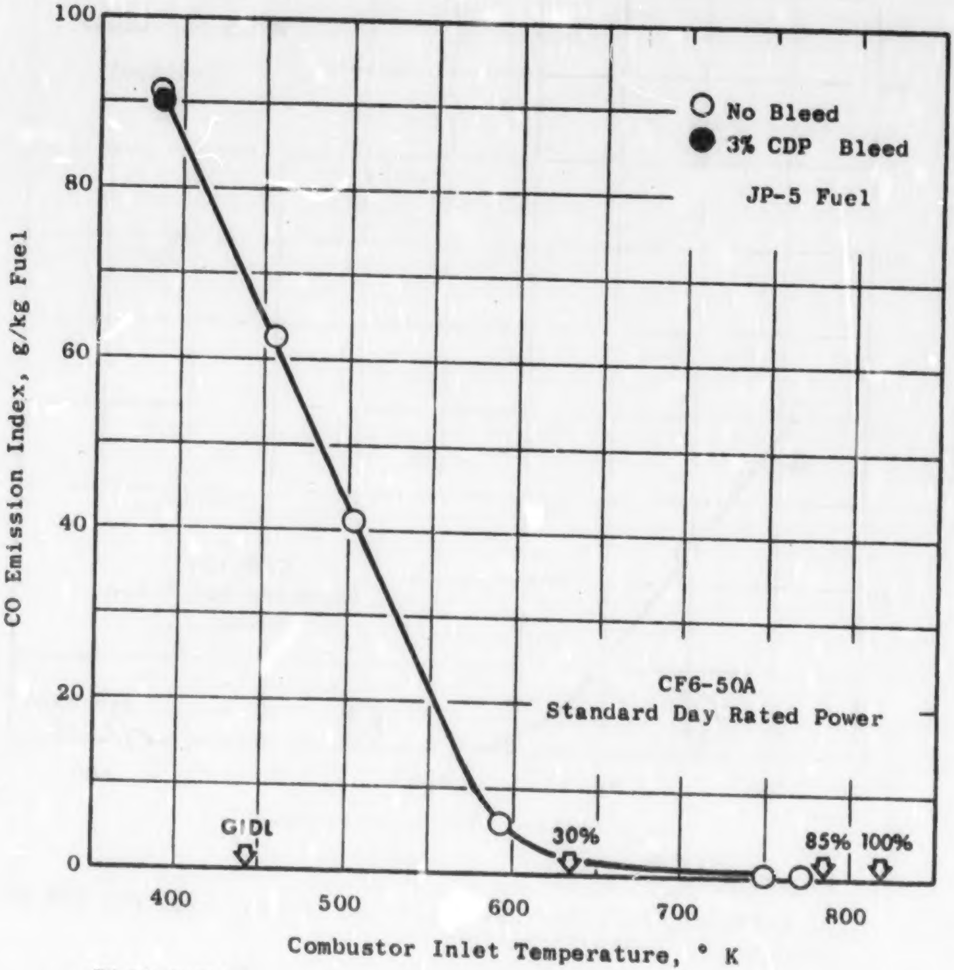


Figure 4.11[39]

CO Emission Characteristics for the General Electric CF6-50 Production Engine/Combustor

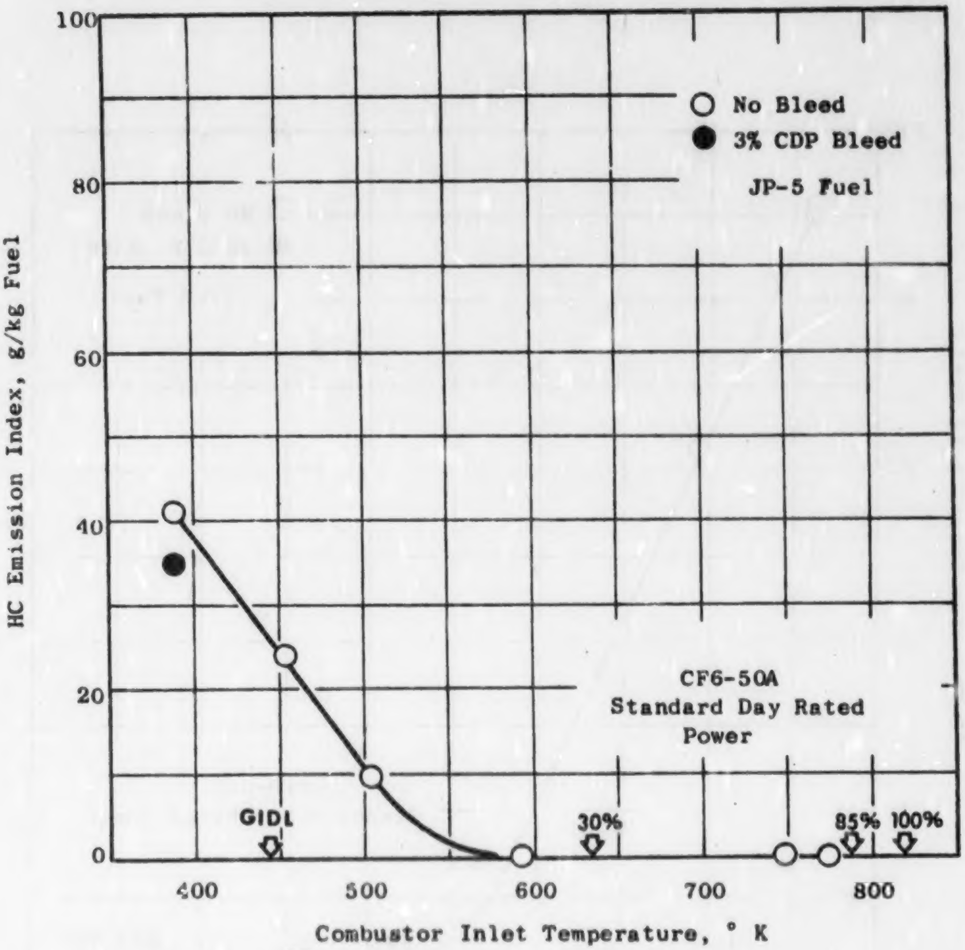


Figure 4.12[40]

HC Emission Characteristics for the General Electric CF6-50 Production Engine/Combustor

4.5 NO_x EMISSIONS

As discussed in section 3.2.2.4, nitrogen oxides produced by combustion of fuels in stationary gas turbines are formed by the combination of nitrogen and oxygen in the combustion air ("thermal NO_x") and from the combination of nitrogen in the fuel with oxygen from the combustion air ("organic NO_x"). These two forms of NO_x contribute to the total NO_x emissions from a given turbine. The variables which affect NO_x emissions from gas turbines have also been discussed in section 3.2.2.4.

4.5.1 Control of Thermal NO_x

It is generally recognized that there are four basic techniques to reduce the formation of thermal NO_x. These are:[41, 42, 43, 44, 45, 47, 48]

a. Reduce the combustion pressure

b. Decrease the peak flame temperatures in the combustor reaction zone

c. Reduce the effective residence time during which the combustion gases remain at elevated temperatures

d. Control the amounts of nitrogen and oxygen available for the production of NO_x

Most control techniques, therefore, accomplish NO_x reductions by designs which incorporate one or more of these techniques.

4.5.1.1 Control by Water or Steam Injection (Wet Techniques) - In section 3.2.2.4.1, it was shown that the formation of thermal NO_x is extremely sensitive to flame temperature . . . increasing exponentially with increases in flame temperatures. Wet control techniques involve the injection of water or steam into the combustion process. The injected fluid provides a heat sink which absorbs some of the heat of reaction thereby reducing peak combustion temperatures and the rate of NO_x formation.

The degree of NO_x reduction achieved for a given turbine depends on the rate and method of introducing the water. In experiments to maximize effectiveness, manufacturers have found that direct injection of atomized water into the primary zone of the combustor is best.[49, 50, 51, 52] Figure 4.13 presents a summary of NO_x emission data from gas turbines using wet control techniques. The effectiveness of wet control techniques in reducing NO_x emissions is illustrated in Figure 4.14 where reductions of NO_x emissions in excess of 80 percent are shown. The band on Figure 4.14 is a compilation of information provided by three major manufacturers of gas turbines, with the spread being due primarily to differences in combustor design, type of fuel used and method of water or steam injection. Data points superimposed on the band represent the results of tests by EPA, operators of gas turbines, and other manufacturers. Although one might expect that water injection would be more effective due to the latent heat of vaporization, many investigators have found only minor differences between water and steam injection. This is explained by the superior mixing characteristics of steam and the finite evaporization time of the liquid droplets.[54]

The industry readily accepts the fact that water and steam injection are valid techniques for reducing NO_x emissions from gas turbines. Turbodyne routinely guarantees an NO_x emission limit of 75 ppm at 15 percent stack gas oxygen and Turbo Power and Marine currently offers water and steam injection on an "as required basis".[55, 56] General Electric states that water or steam injection may be applied to any machine in the product line, that water injection is currently applied to, some MS5000 and MS7000 series machines, and that steam injection has been applied to the MS5000 series turbines for load augmentation on industrial machines since 1961.[57] General Electric also states that steam injection does not appear to affect turbine life and an assessment of the effect

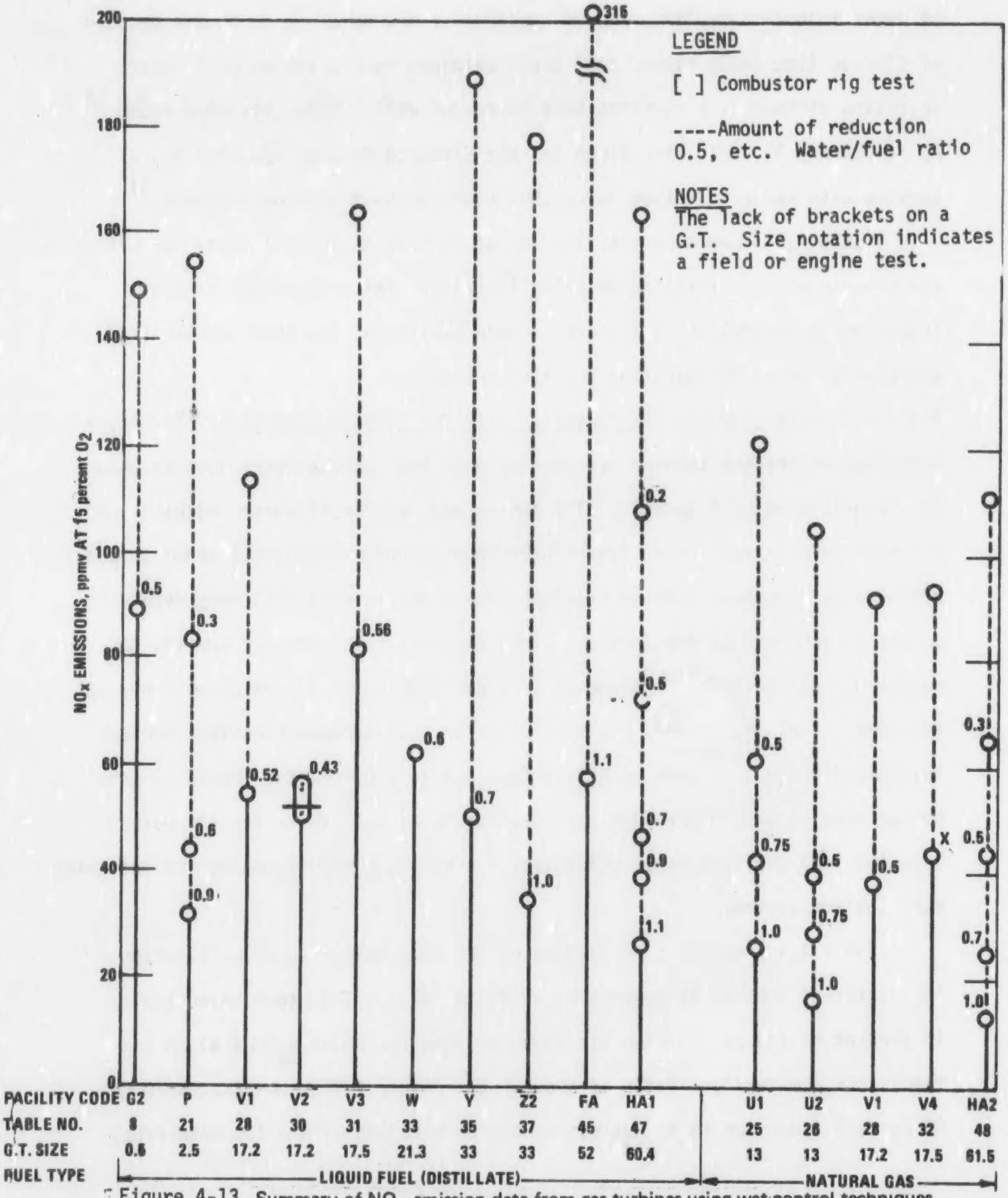


Figure 4-13. Summary of NO_x emission data from gas turbines using wet control techniques.

of water injection on life, and maintenance is not possible to obtain because of limited time (4000 hours) that their machines have operated with water injection although the machines have performed well within this time frame.[58] As of January 31, 1975, San Diego Gas and Electric Company operated a gas turbine with water injection for 17,951 hours without turbine problems.[59]

Table 4.2 shows a partial listing of 74 turbine installations for utilities and process plants, existing or being installed, which use water or steam injection to control NO_x emissions. Where available, the hours accumulated using water or steam injection are also noted.

4.5.1.1.1 Water Quality for Injection into the Turbine Combustor - To reduce corrosion of the gas turbine components, only high purity water must be used for long-term water injection. The amount and quality of water injected into any turbine will vary from manufacturer to manufacturer and from model to model produced by a company. Water treatment requirements will also vary depending on the fuel burned by the turbine, since manufacturers commonly specify the combined total quantity of elements in fuel, water, and air which must not be exceeded. Table 4.3 summarizes typical water quality specifications for gas turbines and compares them to boiler feed water requirements. Obviously the boiler feed water requirements are more stringent than those for the water injected into gas turbines. Utilities, therefore, commonly use equivalent water purification systems.

Table 4.4 presents a comparison of the consumptive water utilization for different schemes of generating 60 MW of power. The steam injection at 15 percent of the gas turbine air flow was used for power augmentation and represents a water/fuel ratio of about 7.5. Figure 4.14 shows that a 70 to 90 percent reduction in NO_x emissions can be obtained with a 1.0 water/fuel

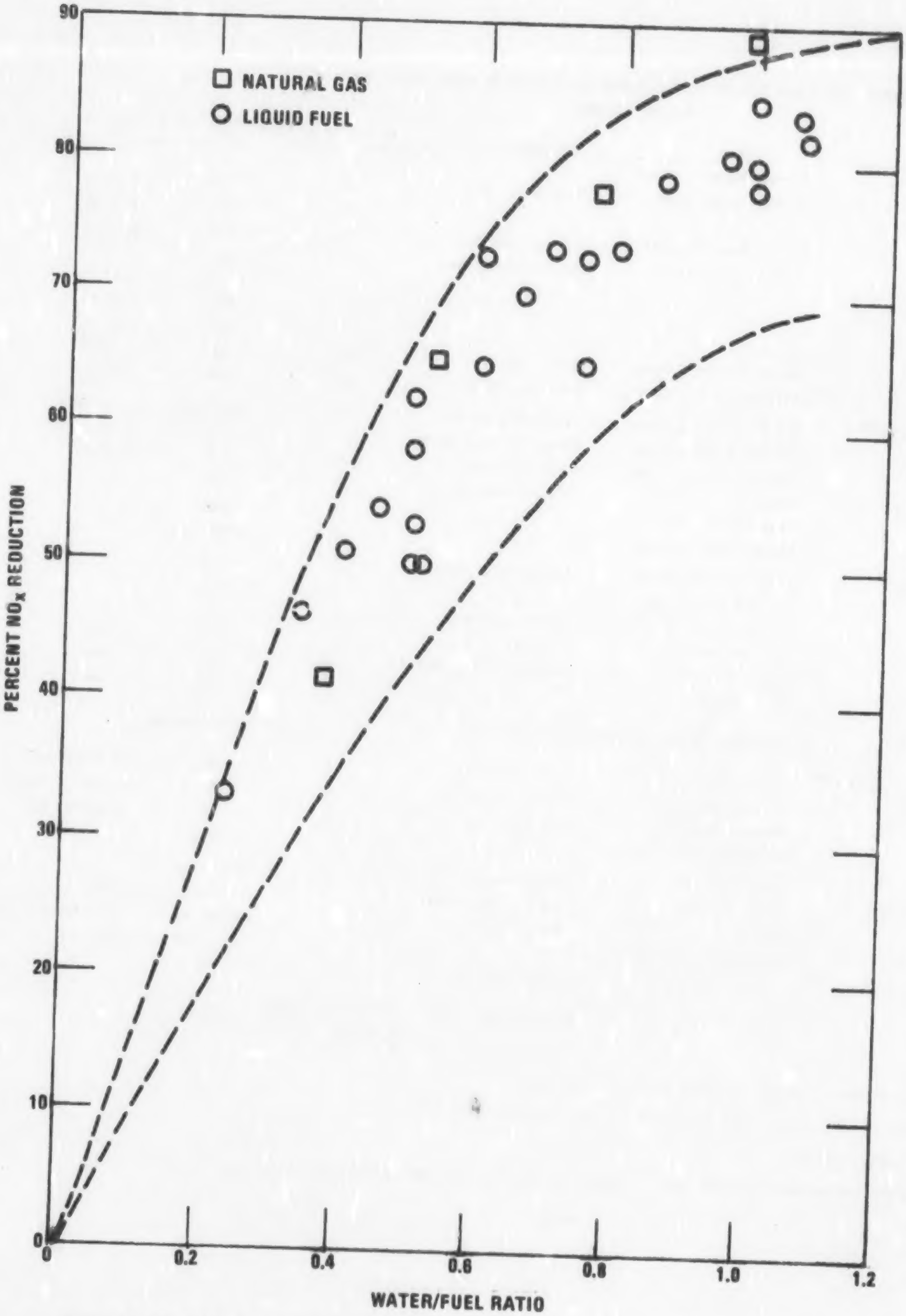


Figure 4.14. Effectiveness of water/steam injection in reducing NO_X emissions.

TABLE 4.2 KNOWN TURBINE INSTALLATIONS WITH WATER OR STEAM INJECTION FOR NO_x CONTROL

	Manufacturer	User	Location	No. of Turbines	Accumulated Hours	
					HRS	Date
WATER	Westinghouse[60]	Florida Power & Light		4(1)	---	---
		Florida Power Corp.	Enterprise, Fla.	1	718	8-31-75
				1	614	8-31-75
		Kansas Power & Light	Abilene, Kansas	1	3194	8-31-75
		Jacksonville Elect	Jacksonville, Fla.	1	857	
				1	1233	7-31-75
				1	1008	7-31-75
				1	779	7-31-75
		Kansai Electric Power	Osaka, Japan	NA	NA	NA(3)
		(2) Southern Calif. Edison	Daggett, Calif.	NA	NA	NA
	General Electric[61]	San Diego Gas & Elect	San Diego, Calif.	17	1000/unit	10-31-75
		San Diego Gas & Elect	Naval Training Center	1	17,951	1-19-75
		Houston Power & Light	Houston, Texas	1	100	10-31-75
		Tucson Gas & Elect	Tucson, Arizona	1	100	10-31-75
		Iowa Public Service	---	3	500	10-31-75
		Kansas Power & Light	---	3	3000 Total	10-31-75
	GE - ON ORDER[62]	City of Jacksonville	Jacksonville, Fla.	1	---	---
		Florida Power Corp.	---	4	---	---
		Portland General Elect.	Portland, Oregon	6	---	---
		Arizona Power & Light	---	1	---	---
		(2) Southern Calif. Edison	Lucerne Valley, Calif	---	---	---
		Ohio Edison	---	1	---	---
	Turbodyne[63]	Houston Lighting & Power	Houston, Texas		not installed yet	
	Turbo Power & Marine[64]	Burbank Public Service	Burbank, Calif.	1	NA	installed 4/74
		City of Glendale	Glendale, Calif.	1	NA	installed 10/73
		City of Pasadena	Pasadena, Calif.	2	NA	installed 5/74
		Southern Calif. Edison	Goleta, Calif.	2	27.3 each	1-13-76
STEAM	Westinghouse[65]	Union Carbide	Texas City, Texas	1	3000 plus(4)	
	General Electric[66]	Union Carbide	Texas City, Texas	1	15 years - June 1975	
		Exxon	Baytown, Texas	7	48,000 as of 2-15-73 on one turbine	
	Turbodyne[67]	Southern Calif. Edison		7	not installed yet	
	Turbo Power & Marine[68]	City of Pasadena	Pasadena, Calif.	1	NA	NA

TOTAL INSTALLATIONS: 58 MINIMUM, WATER
16 STEAM

NOTES:

(1) Water injection system available but not being used.

(2) 19 gas turbines using water injection - total for both facilities

(3) NA - not available

(4) Unit was overhauled at 47,000 hrs. of operation with the last 3000 using steam injection.

Table 4.3[69,70,71,72,73]

Water Quality Specifications

	Turbine	Boiler Feed
Total Dissolved Solids (TDS) + Non Dissolved Solids (PPM)	1.0 - 5.0	0.25
Sodium + Potassium (PPM)	0.5(1)	0.25
Silica (PPM)	0.02	0.0
Particle Size	10	---
Ph	7.0 - 8.5	6.5 - 7.0

(1) Turbo Power & Marine limits sodium to 0.1 PPM

Table 4.4[74]

Comparison of Consumptive Water Utilization for Different Schemes of Generating 60 MW of Power

Plant Type	Water Utilization lbs/kw-hr
Nuclear (33% efficiency)	6.5 - 6.8
Fossil (40% efficiency)	4.2 - 4.5
Stag combined cycle (38% efficiency)	2.0 - 2.1
Stag combined cycle with steam injection at 15% of turbine air flow	3.4
Simple and Regenerative Cycle Gas Turbine	0

NOTE: (1) 3.4 - 2.0 = 1.4 lb/kw-hr used for steam injection @ 7.5 water/fuel ratio

(2) $\frac{1.4}{7.5}$ = 0.18 lb. of water/kw-hr at a 1.0 water/fuel ratio

(3) $\frac{4.2}{0.18}$ = 23.3 times as much water used by boiler as for a turbine at 1.0 water/fuel ratio

ratio. Using the figures from Table 4.4 and assuming a 1.0 water/fuel ratio, the consumptive water required for control of NO_x emissions from the turbine is about 5 percent of the consumptive water use of a comparable steam boiler. Since the water requirements for control of NO_x emissions are small, they could readily be handled by water transport and storage if water was not available at a site, thereby increasing siting flexibility.[75]

Figure 4.15 shows a water purification system as typically used for gas turbines using water injection. The figure shows flowrates for a water treatment system which could be used to provide water injection for operating five 28 MW gas turbines 10 hours per day. The typical system may use a series of filters followed by a reverse osmosis unit (RO) and a deionization (DI) or demineralization unit. These are further described as follows:[80]

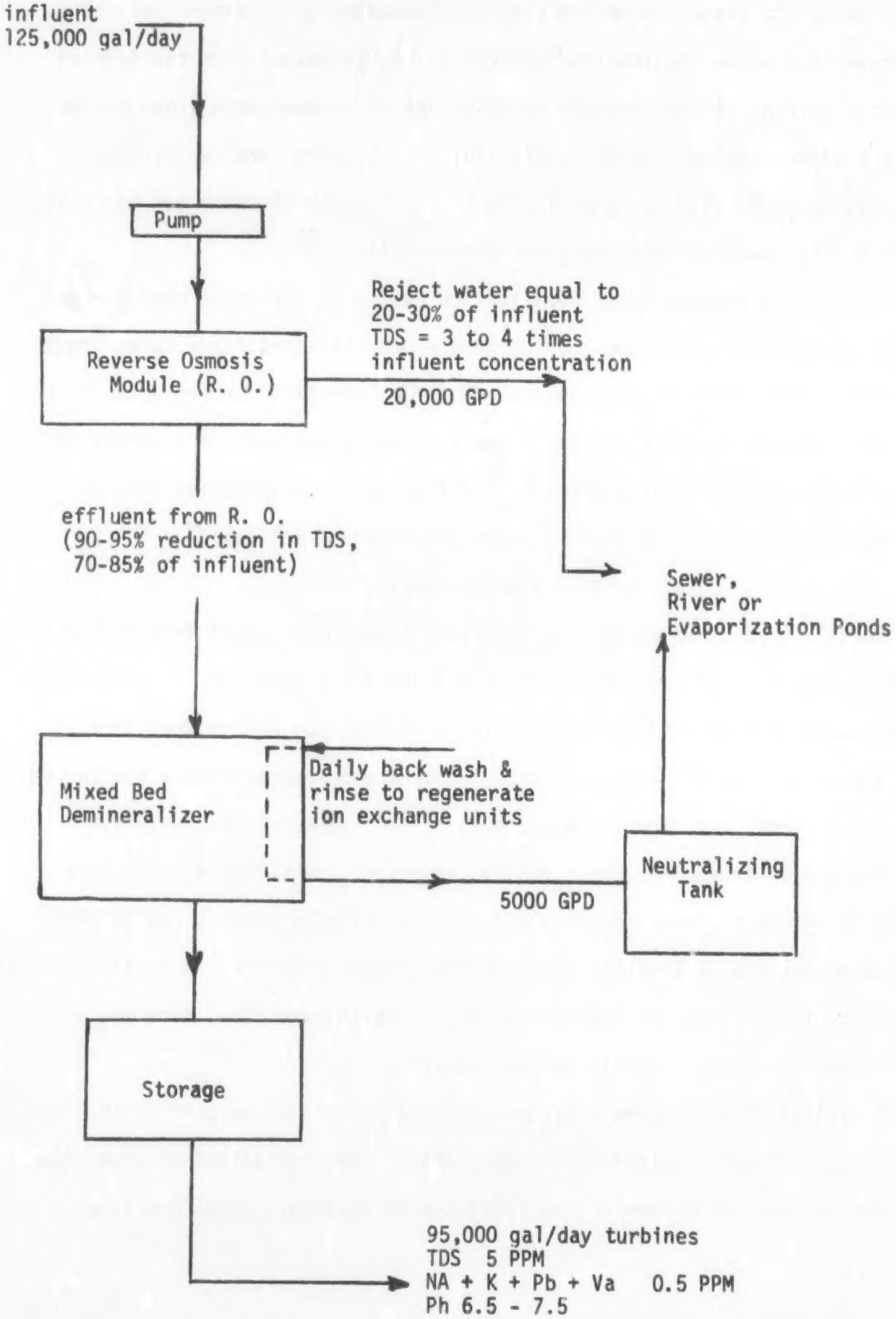
a) Reverse Osmosis (RO)

Osmosis is defined as the spontaneous passage of a liquid from a dilute to a more concentrated solution across a semipermeable membrane - that is, a membrane permeable to one component of the solution, but not to the other.

When solutions of different concentrations are separated by a semipermeable membrane, solvent (water) will permeate to the more concentrated solution. This flow may be stopped or the direction reversed with sufficient pressure applied to the more concentrated solution. The pressure which produces zero net flow across the membrane is known as the osmotic pressure. When the pressure applied is greater than the osmotic pressure, the solvent flows from the more concentrated solution. This is reverse osmosis.

A typical reverse osmosis system can convert 70 - 90 percent of the feedwater to pure product, depending on application. Concentrations of more than 2,000 ppm of total dissolved solids (TDS) can be handled. Inlet pressures of

Figure 4.15[76,77,78,79]
Typical Water Purification System for Five Gas Turbines (28 MW each) Sized to Operate the Turbines 10 Hours Per Day


influent
125,000 gal/day
Pump
Reverse Osmosis Module (R. O.)
Reject water equal to 20-30% of influent TDS = 3 to 4 times influent concentration
20,000 GPD
effluent from R. O. (90-95% reduction in TDS, 70-85% of influent)
Sewer, River or Evaporization Ponds
Mixed Bed Demineralizer
Daily back wash & rinse to regenerate ion exchange units
Neutralizing Tank
5000 GPD
Storage
95,000 gal/day turbines
TDS 5 PPM
NA + K + Pb + Va 0.5 PPM
Ph 6.5 - 7.5

400 - 600 psig are normally employed as the driving force. The product water will be 85 - 95 percent lower in dissolved ions compared to feedwater concentrations. Particulate matter will also be removed. Some materials such as iron or high levels of particulates can cause problems with reverse osmosis systems. To reduce operating costs, many reverse osmosis systems include a series of filters and pretreatment systems.

b) Deionization (DI) or Demineralization

Demineralization is a process of removing the mineral salts from water by ion exchange. Natural waters, whether obtained from wells, rivers, lakes, or even collected rain water, contain some dissolved salts, silica, and carbon dioxide. The salts are ionized and are present as metallic cations (Ca++, Mg++, Na+) and anions (Cl^-, SO_4^{--}, HCO^-_3). The silica and carbon dioxide are dissolved, but ionized to a considerably lesser degree.

There are two basic steps in DI. In the primary stage, a hydrogen cation exchange resin converts the salts to the corresponding acid by exchanging hydrogen ions (H+) for the metallic cations (Ca++, Mg++, Na+). After passage through this acid regenerated resin bed, the water contains a mixture of acids corresponding in concentrations to the anions originally present in the waters. These are most commonly carbonic acid (bicarbonates), hydrochloric acid (chlorides), sulfuric acid (sulfate) and silicic acid (silica). For example, $H_2Z + CaSO_4 \rightleftarrows CaZ + H_2SO_4$.

The second phase consists of removing these acids by an alkali-regenerated anion exchange resin. The most common alkali used is caustic soda (sodium hydroxide). The resin may be either of two types:

(1) A "weakly basic" anion exchange resin which removes only strong "mineral acids", such as hydrochloric, sulfuric and nitric, etc., and permits "weak" carbonic and silicic acids to pass through.

(2) A "strongly basic anion exchange resin which removes "weak" carbonic and silicic acid as well as "strong" mineral acids, replacing the chloride, sulfate, and bicarbonate anions with hydroxyl (OH^-) ions. When carbon dioxide and silica removal is desired, a strong base anion resin is used. Where their removal is not required, a weak base anion resin can be used. When mineral acids in the cation effluent are high enough to justify the additional investment in resin and equipment, a system consisting of a cation unit, a weak base anion unit, and a strong base anion unit in series can be used.

In practice, RO by itself will not be adequate to achieve the quality of water required for injection. RO can, however, act to significantly reduce the load on the DI unit, allowing these units to be smaller and require less chemicals for regeneration.

A choice exists in regard to the configuration of the DI unit. This choice involves the decision whether a dual bed deionizer should be used with separate cation and anion exchange resins or whether a mixed bed system should be used. Also, some companies feel that the dual bed should be used in combination with the mixed bed system. The dual bed, followed by a mixed bed, has a higher capital cost but lower operating costs since less frequent regeneration is required.

4.5.1.1.2 The Effect of Water or Steam Injection on Emissions of Particulates, CO, and HC - The data reported in the literature is contradictory regarding the effect of water or steam injection on the emissions of particulates, CO and HC. In tests performed by Westinghouse on the model W-251 gas turbine, water injection resulted in slightly lower HC emissions at base and peak loads, about a 75 percent reduction in CO emissions at base load, and a reduction in visible emissions by 1.5 to 1.75 smokespot numbers (ASTM-D-2156).[81] Data reported from tests of the New York Power Pool turbines resulted in the conclusion that the general effect of water injection on a Turbo Power and Marine FT4 turbine is to reduce CO, HC, and particulate emissions while the turbine operates at base load but to increase these emissions when the turbine operates at peak load.[82] Tests performed by General Motors show that the effect of water injection on CO is a function of the water injection method and that NO_x reductions can be obtained without CO increases.[83]

Therefore, it appears that the effect of water or steam injection on emissions of HC, CO, and particulates is related to the water injection method and increases in these pollutants can be avoided by proper design of the injection system.

4.5.1.1.3 The Effect of Water or Steam Injection on the Conversion of Fuel Bound Nitrogen to NO_x (Organic NO_x) - As discussed in Section 3.2.2.4.2, the exact mechanism for formation of organic NO_x is not known, and organic NO_x is only a problem when burning residual oils, some crude oils, or heavy distillate fuels which have high nitrogen contents. Section 3.2.2.4.2 also shows that most light distillate fuels contain less than 0.015 percent by weight nitrogen, crude oils generally contain less than 0.2 percent, and residual oils can go as high as 2 percent.

Data concerning the effect of water injection on the conversion of fuel bound nitrogen is scarce. General Electric has performed some very limited tests on a combustor rig using "nitrogen doped" fuels and by using this data in conjunction with some field test data have extrapolated the results to predict NO_x emissions from an MS-7001C simple cycle turbine burning fuels with different nitrogen contents at three firing temperatures as shown in Figure 4-16.[84] The water to fuel ratio has been approximated and added to the figure by assuming a fuel to air ratio of 0.015, 0.018, and 0.021 at firing temperatures of 1600°F, 1800°F, and 2000°F, respectively.

The curves presented in Figure 4.16 represent total NO_x emissions - thermally formed NO_x plus organic NO_x. Looking at the curves for the 0.5 percent nitrogen fuel, one can observe that beyond a water to fuel ratio of about 1.2 to 1.3, the injection of additional water results in increases in emissions of NO_x. This would indicate that, beyond certain high rates of water injection, an increased yield of organic NO_x may result. These water injection rates, however, are substantially greater than would commonly be used for NO_x control, since a 1.0 water/fuel ratio will provide 70 to 90 percent reduction in thermal NO_x as explained in section 4.5.1.2. Also, some of the water injection rates shown in Figure 4.16 would exceed the levels considered as a maximum for an operating gas turbine because the flame would be extinguished.[86]

4.5.1.1.4 The Effect of Water and Steam Injection on Gas Turbine Efficiency - Water and steam injection for control of NO_x are reported to effect gas turbine efficiency. Table 4.3 summarizes the data reported by manufacturers and users of gas turbines in response to queries by EPA. The data reveal that water injection reduces gas turbine efficiency by about one percent for a water to fuel ratio of one. If waste steam is available from a boiler, steam injection at a water to fuel ratio of one results in about a one percent increase in gas turbine efficiency. For a combined cycle turbine using steam from the boiler portion, there is an overall reduction in efficiency of one percent at a 1.0 steam/fuel injection rate.1/

4.5.1.1.5 The Effect of Very High Water or Steam Injection Rates on Mass Emissions of NO_x - Figures 4.17 and 4.18 show calculated NO_x reductions as a function of water and steam injection rates, respectively. In Figure 4.14, it can be observed that the effectiveness of water or steam injection in reducing NO_x emissions is sharply reduced beyond about a 1.0 water to fuel ratio.

1/A one percent decrease in overall turbine efficiency corresponds to about a 5 percent increase in fuel consumption.

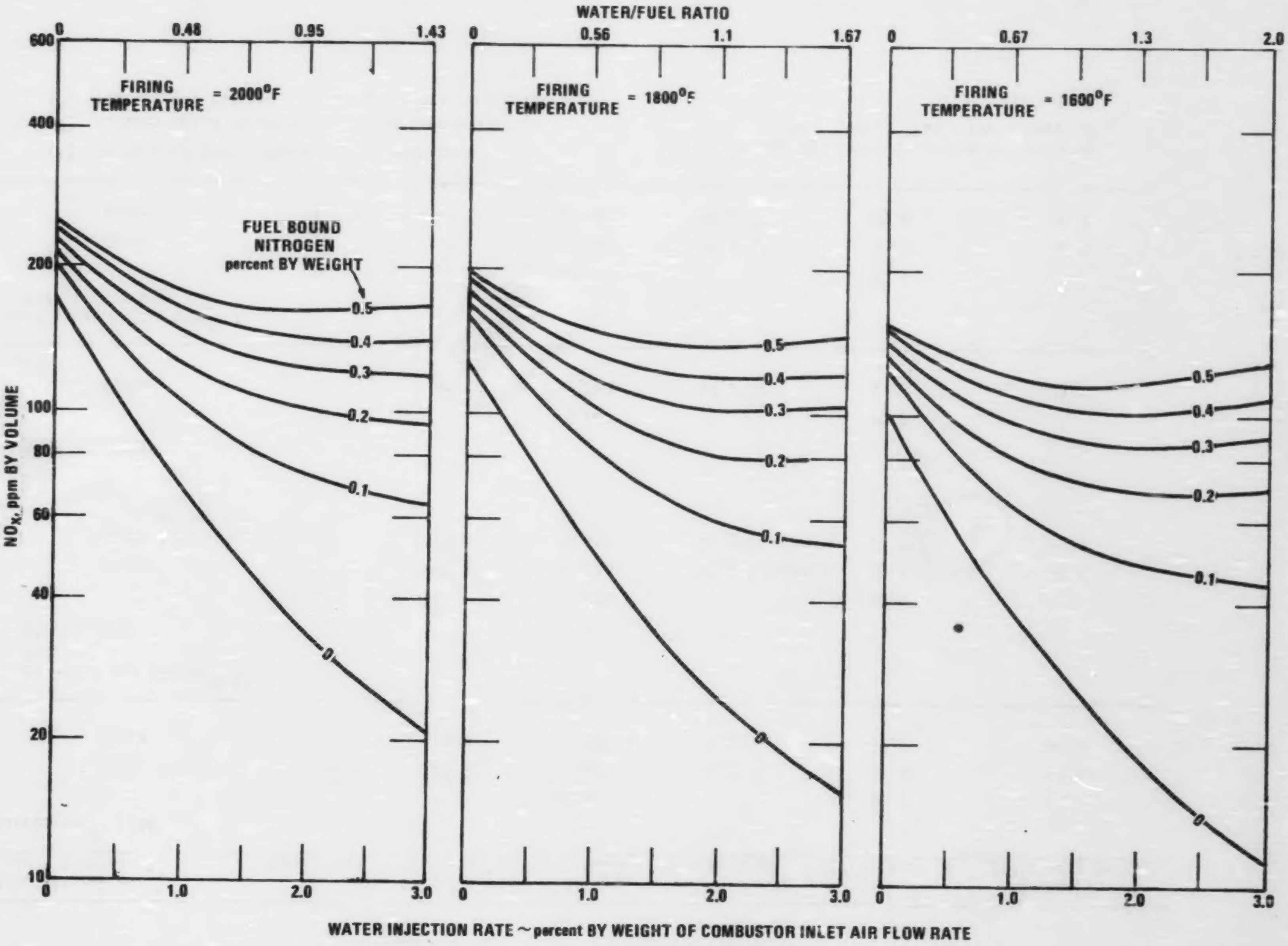


Figure 4.16[85]. Predicted NO_x emissions for a range of nitrogen contents and water injection rates. MS-7001C simple cycle, NEMA conditions.

Table 4.5 THE EFFECTS OF WATER AND STEAM INJECTION ON POWER AUGMENTATION AND GAS TURBINE EFFICIENCY

Manufacturer or User & Turbine		Water/fuel	Percent Base Power Change	Heat Rate Btu/kw-hr	Percent Change in Heat Rate	Efficiency %	Change in Efficiency Percent
Turbodyne[87] 11D4							
		0		11,250		30.3	
	Water	1.3(1)	+9.6	11,840	+5.2	28.8	-1.5
	Steam	1.3(1)	+9.9	10.790	-4.0	31.6	+1.3
Turbo Power and Marine[88] 1/ FT4C							
Simple Cycle							
		0		11,040		30.9	
	Water	1	+8.7	11,260	+2.0 augmented	30.3	-0.6
	Steam (waste heat boiler for turbine only)	1	+8.0	10,930	-1.0(2)	31.2	+0.9
Combined cycle							
		0		8,007		42.6	
	Steam	1	+0.86	8,263	+3.2(3)	41.3	-1.3
Pacific Gas & Electric[89] FT4C-1 Twin Pac							
		0		11,040		30.9	
	Water	1	+7.1	11,282	+2.2	30.3	-0.6
	Steam	1	+7.1	11,460	+3.8(3)	29.8	-1.1

(1) calculated from information in reference

(2) assumes waste steam is available from boiler

(3) overall gas turbine - steam cycle heat input increase

1/ May 28, 1976, telephone conversation, Glenn Edgerly, TP&M - Ken Durkee, EPA

The combined result of the negative effect of water injection on efficiency (as discussed in 4.5.1.1,4) and the maximum NO_x reduction at about a 1.0 water to fuel ratio is an actual increase in the mass emission rate of NO_x with additional water. As shown in Figure 4.17, this phenomenon occurs at water to fuel ratios of about 1.4 and 1.9, and above.

If one assumes that waste steam is available for injection, this phenomen does not occur - as shown in Figure 4.18. If the affect on efficiency of producing the steam is included, the phemonmenon will occur as in Figure 4.17.

4.5.1.1.6 The Effect of Water or Steam Injection on the Formation of Ice Fog - Ice fog occurs only under certain atmospheric conditions and is a serious local problem in only a small portion of the United States, primarily Alaska. Ice fog occurs at temperatures below about -20°F and consists of highly reflective, small (mean diameter 3 to 7 microns) ice crystals which are nucleated by air-borne particulates.[90] It occurs naturally over lakes and rivers. Man-made ice fog is caused by exhausts of automobiles and power plants, etc.

Ice fog severely restricts visibility since the crystals are long-lived. Once it has formed, it may continue to plague auto and air traffic for extended periods. The formation of ice fog is difficult to predict.

Test data are not available on the operation of gas turbines in artic areas, so the actual impact of water or steam injection is unknown. However, water or steam injection will significantly increase the moisture content of the exhaust gases. Some obvious, but unanswerable questions, are:

(1) Will the exhaust gases from a gas turbine penetrate an inversion layer and escape to the upper atmosphere?

(2) What are the climatological conditions which will permit formation of ice fog in the exhaust of gas turbines?

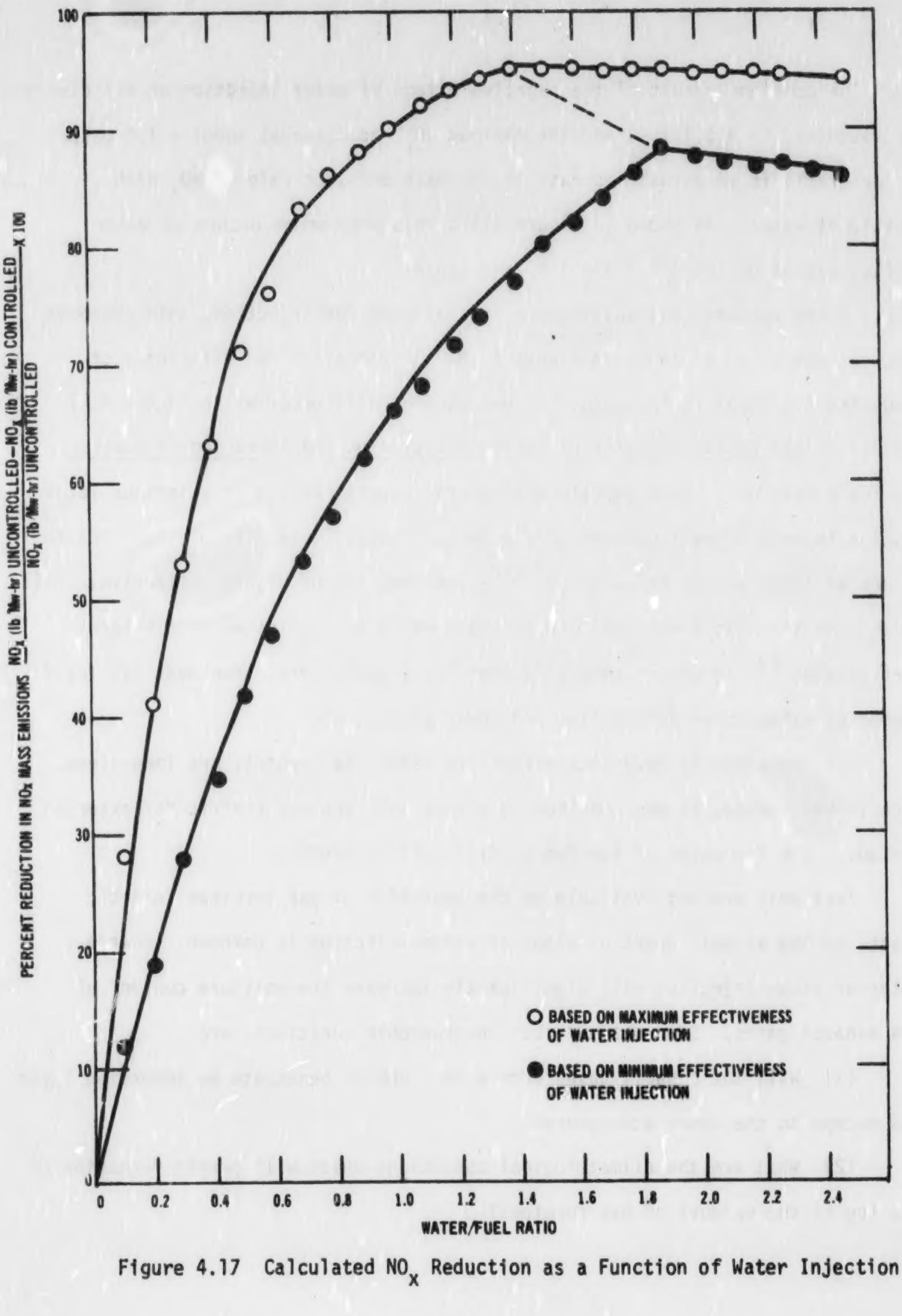


Figure 4.17 Calculated NO_x Reduction as a Function of Water Injection Rate

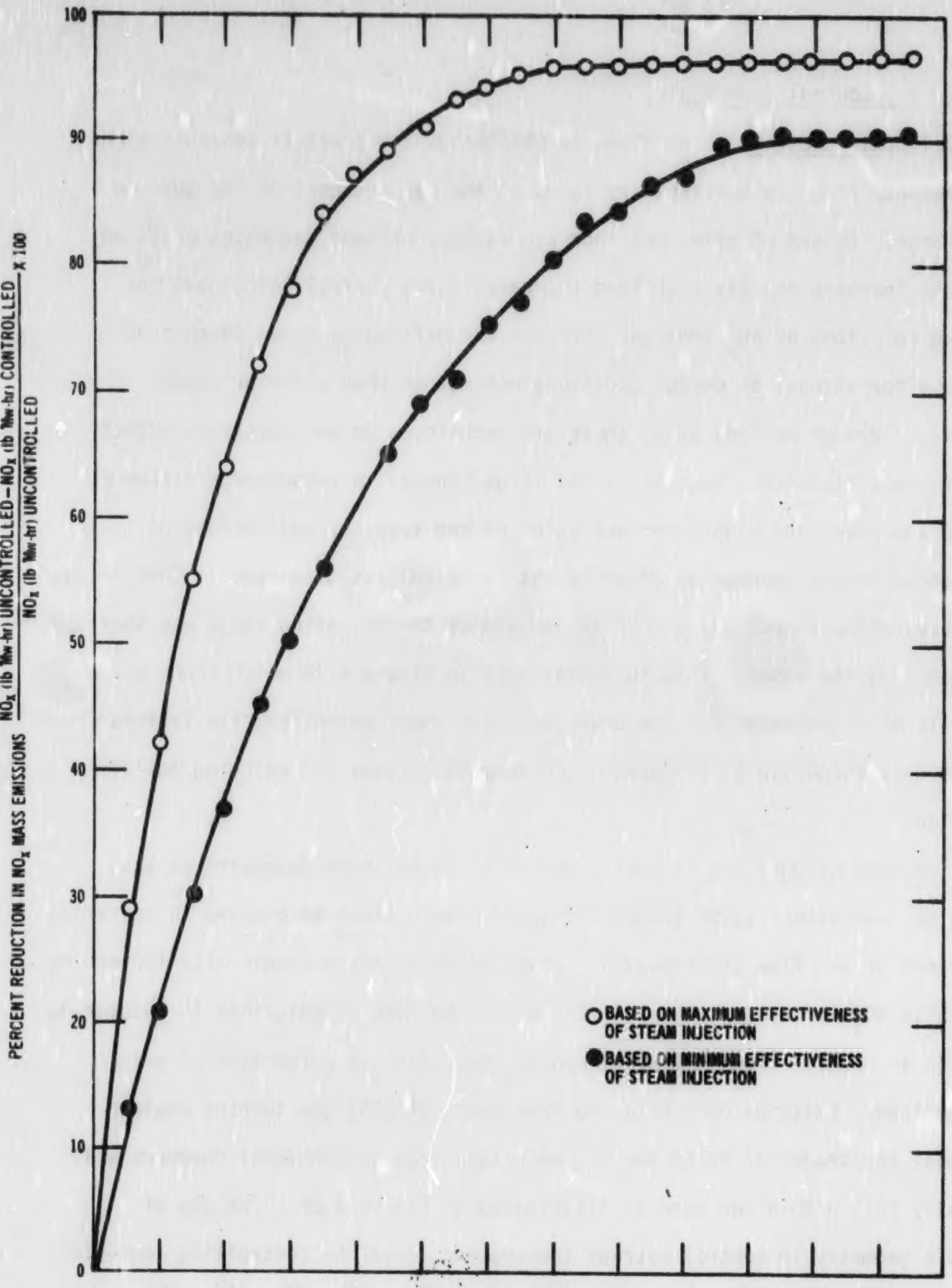


Figure 4.18 Calculated NO$_x$ Reduction as a Function of Steam Injection Rate

4.5.1.2 Dry Control Techniques

4.5.1.2.1 Introduction - As shown in Chapter 3, the greatest concentrations of pollutants from gas turbines are found at the two extremes of the turbine power range. CO and HC emissions increase rapidly as load decreases while NO_x emissions increase rapidly with load increases. Dry control techniques for reducing emissions of NO_x from gas turbines are defined as those techniques which use operational or design modifications rather than water or steam injection. Proper designs using these dry techniques do not appear to affect gas turbine efficiency. Because of the large changes in combustor environment between low power and high power operation of gas turbines and because of differences in the mechanisms of pollutant formation (as discussed in Chapter 3), techniques which reduce pollutants at one end of the operating range may increase pollutants at the other. This is illustrated in Figure 4.19 which shows a schematic of a combustor for low power and high power operation, the factors which affect formation of pollutants and some techniques for reducing pollutant formation.

For conventional gas turbine combustors, it has been demonstrated that CO and NO_x emissions can be traded off against each other by changes in operating conditions or air flow distribution. Since CO emissions decrease with increasing load while NO_x emissions increase with increasing load as described in Chapter 3, there is an obvious trade-off of CO and NO_x emissions as a function of gas turbine load. Emission data obtained from tests of many gas turbine engines show that the trade-off of CO and NO_x emissions from conventional chambers will generally fall within the band as illustrated in Figure 4.20. The use of variable geometry to control average temperature levels by controlling combustor air distribution will result in a trade-off of CO and NO_x emissions by shifting the emissions performance point either up or down the characteristic combustor

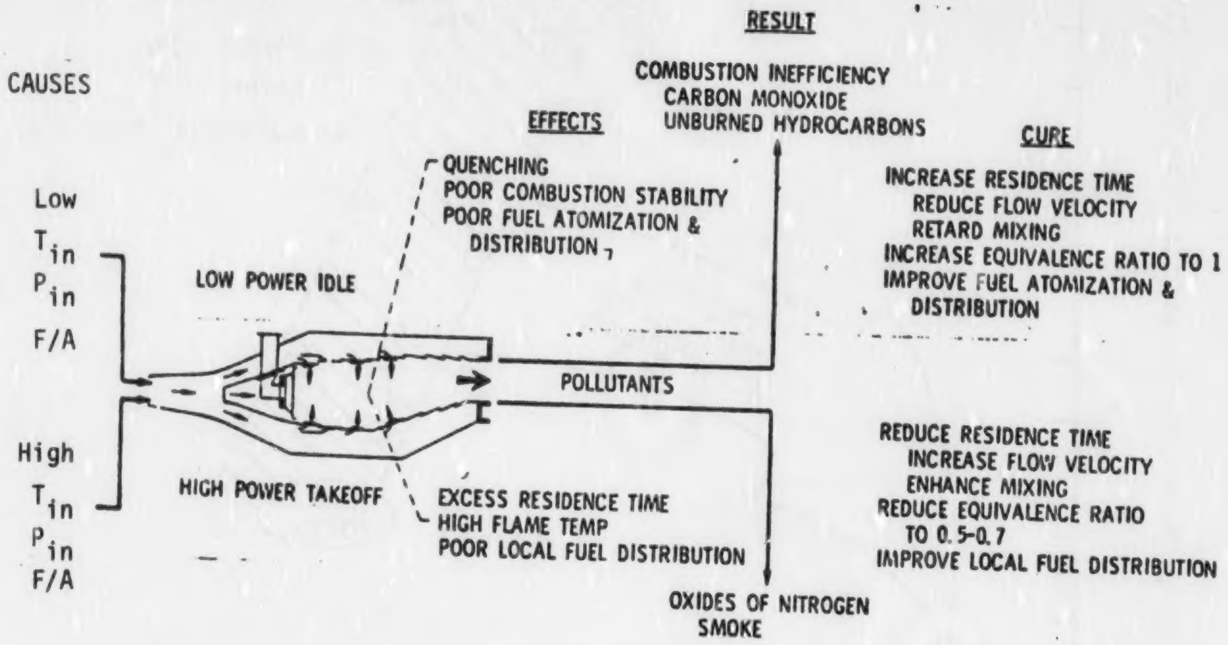

RESULT
COMBUSTION INEFFICIENCY
CARBON MONOXIDE
UNBURNED HYDROCARBONS
CAUSES
EFFECTS
CURE
QUENCHING
POOR COMBUSTION STABILITY
POOR FUEL ATOMIZATION & DISTRIBUTION
INCREASE RESIDENCE TIME
REDUCE FLOW VELOCITY
RETARD MIXING
INCREASE EQUIVALENCE RATIO TO 1
IMPROVE FUEL ATOMIZATION & DISTRIBUTION
Low
T in
P in
F/A
LOW POWER IDLE
POLLUTANTS
High
T in
P in
F/A
HIGH POWER TAKEOFF
EXCESS RESIDENCE TIME
HIGH FLAME TEMP
POOR LOCAL FUEL DISTRIBUTION
REDUCE RESIDENCE TIME
INCREASE FLOW VELOCITY
ENHANCE MIXING
REDUCE EQUIVALENCE RATIO TO 0.5-0.7
IMPROVE LOCAL FUEL DISTRIBUTION
OXIDES OF NITROGEN
SMOKE

Figure 4.19 [91] Factors Effecting the Pollutants From Gas Turbines at Low Power (Idle) and High Power Conditions

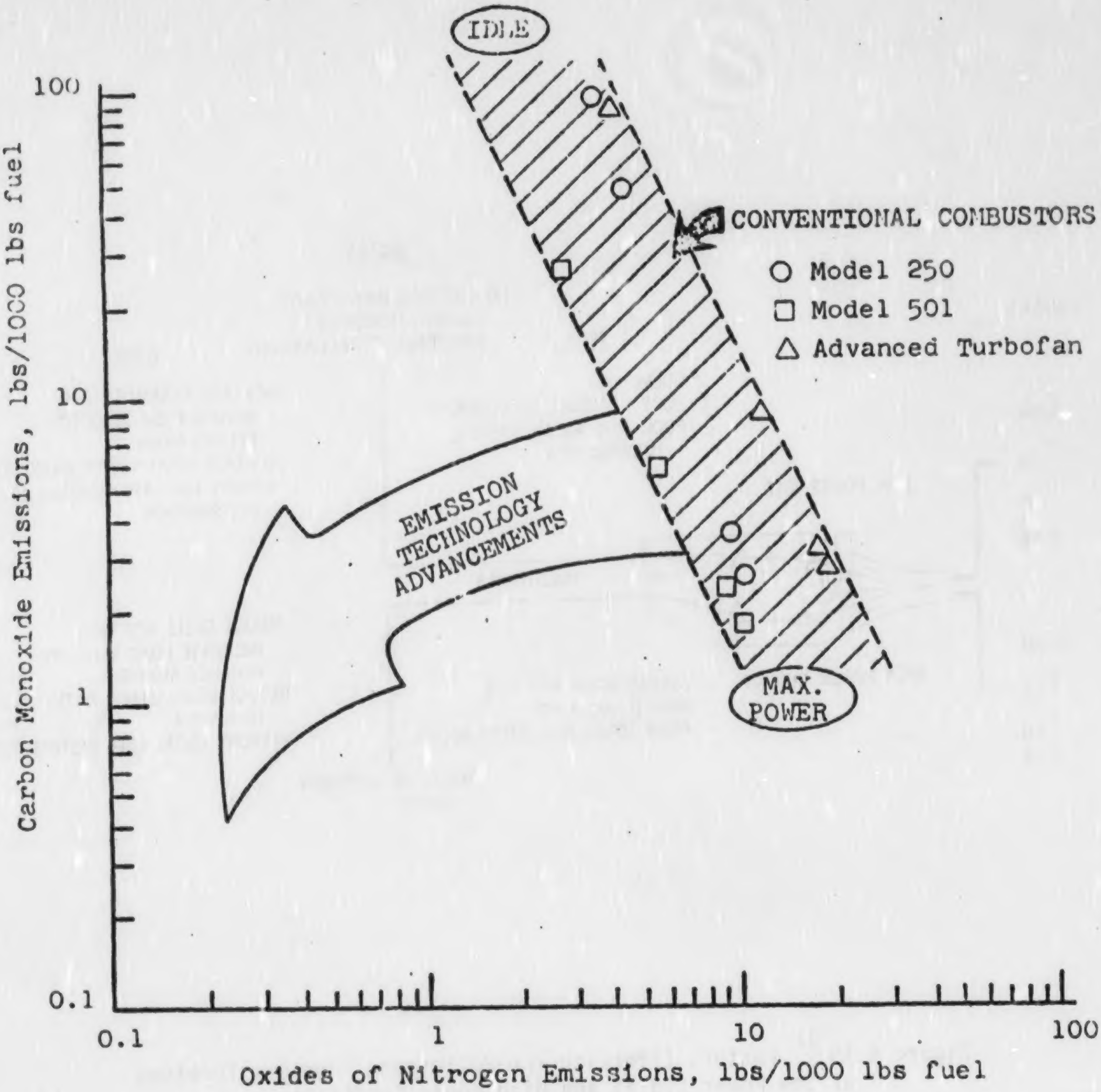


Figure 4.20[92] CO and NO_x Performance of Conventional Gas Turbine Combustors

performance line as shown in Figure 4.21. This shift could be obtained by minor redesign of a given combustor using fixed geometry to obtain a NO_x - CO trade-off at a given operating point but the use of variable geometry may then be required to operate the engine over the full load range.

An undesirable feature of this trade-off is that it merely trades one pollutant for another. Reductions of both pollutants (CO and NO_x) will require new combustor designs which either shift the CO - NO_x curve to the left as shown in Figure 4.20 or change the CO - NO_x characteristic trade-off curve as shown in Figure 4.22 for a prototype prechamber combustor using variable geometry.

Combustor designs utilizing dry control techniques to retard the formation of thermal NO_x usually involve design modifications to influence:

a. the reaction flame temperature
b. residence time of the gases at temperature
c. the amounts of oxygen available for conversion to NO_x
d. atomization and vaporization of the fuel
e. mixing of the fuel and air

The effects of variations in these factors have been discussed in Chapter 3 and, therefore, will not be addressed again. Combustor design modifications which are applied to accomplish the factors listed above include techniques such as:

a. air staging and redistribution
b. fuel vaporization
c. fuel staging
d. two stage combustion and off stoichiometric combustion
e. premixing of the air and fuel prior to introduction to the combustion chamber
f. variable combustor geometry

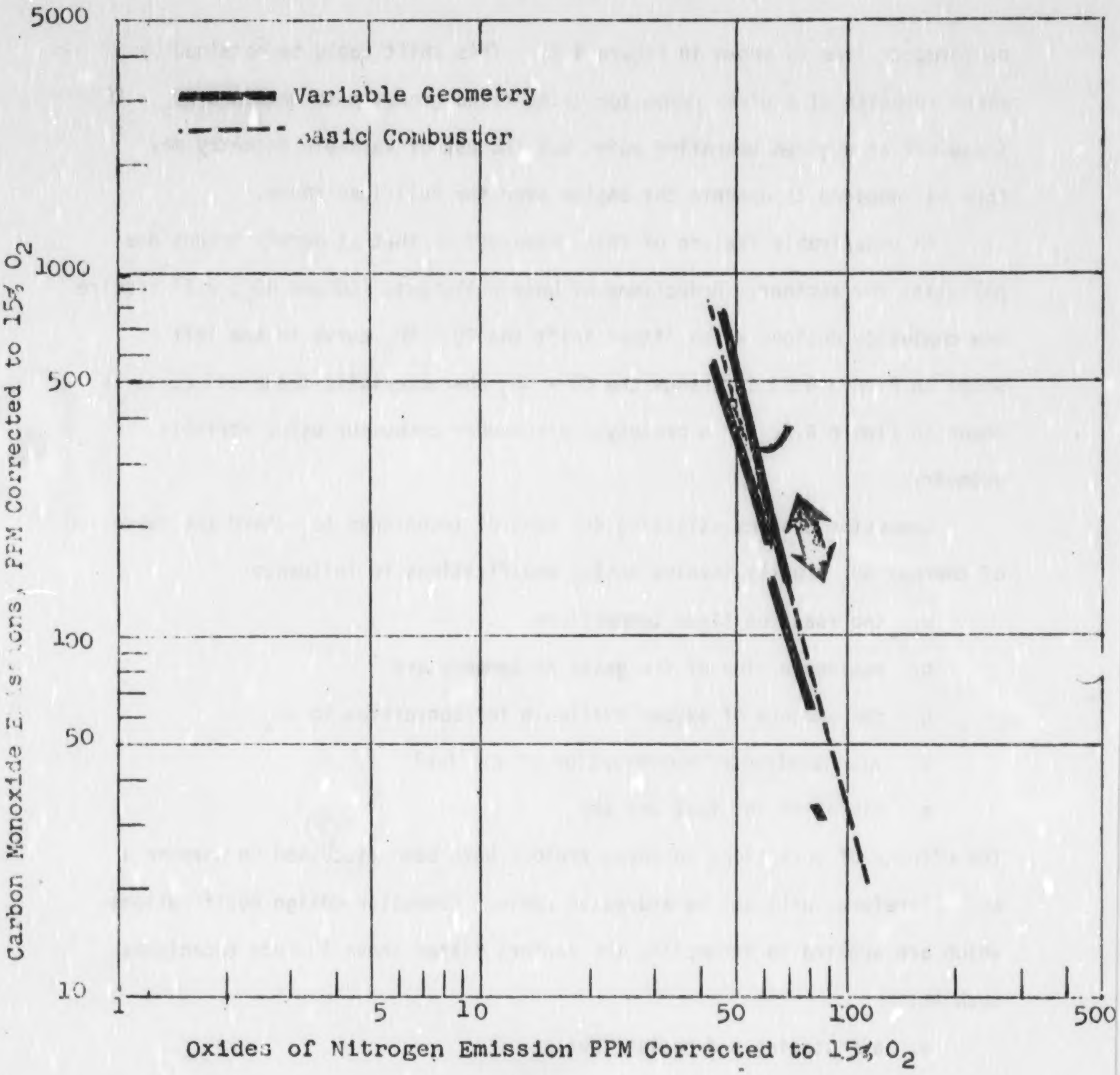


Figure 4.21[93] Effect of Variable Geometry Control on Exhaust Emissions from the T-63 Gas Turbine Engine Combustor

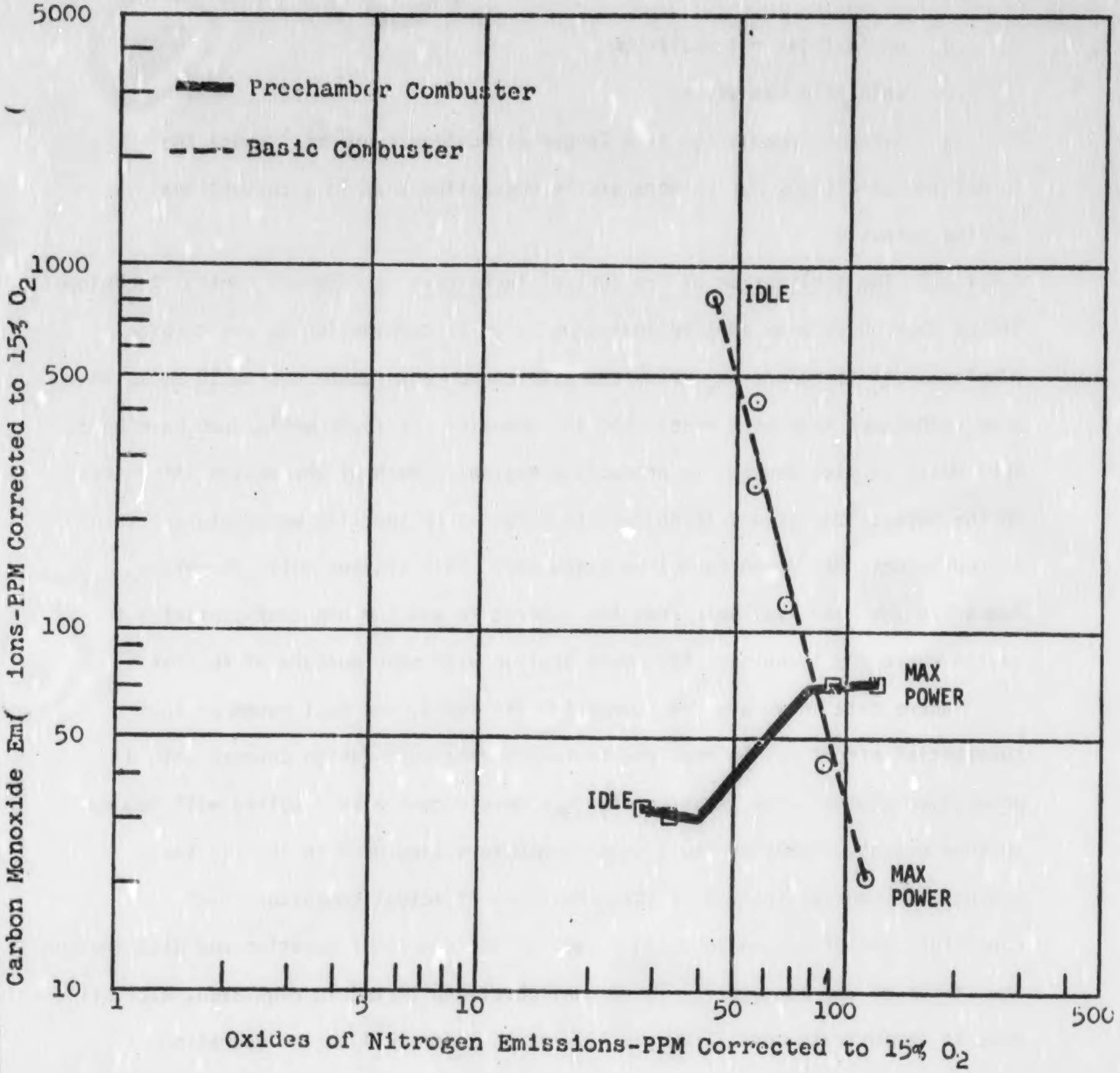


Figure 4.22 [94] CO versus NO_x for Rig Tests of the T-63 Combustor and a Prototype T-63 Prechamber Combustor Using Variable Geometry

g. exhaust gas recirculation

h. catalytic combustion

i. external combustion in a larger combustion chamber(s) where the combustion conditions can be more easily controlled than in a conventional gas turbine combustor

4.5.1.2.2 The application of dry control techniques - The dry control techniques listed above have been applied individually or in combination by gas turbine manufacturers to reduce NO_x emissions with concurrent reductions in CO emissions. Many techniques have been researched in combustor rig tests while some have been applied in varying degrees to production engines. Much of the design information on the application of dry techniques to turbines by specific manufacturers is of a proprietory nature and cannot be discussed. This section will, therefore, summarize the data available from the literature and the non-confidential information where dry techniques have been applied with some measure of success.

Where data shown are from combustor rig tests, one must remember that substantial effort may be required to incorporate such design changes into a production engine. The amount of further development work required will depend to some extent on whether "full size" combustors were used in the rig tests and on the sophistification of the simulation of actual combustor inlet conditions and of engine conditions such as the flow configuration and distribution downstream of the combustor. Those configurations tested in rigs also, will still have to demonstrate durability and life tests under full engine operating conditions.

As shown in Figure 4.23, the use of advanced fuel injection methods such as airblast fuel atomization have been found to reduce NO_x emissions by 10 to 15 percent when compared to the more conventional fuel injection methods. This is primarily due to the better fuel and air mixing which reduces any localized

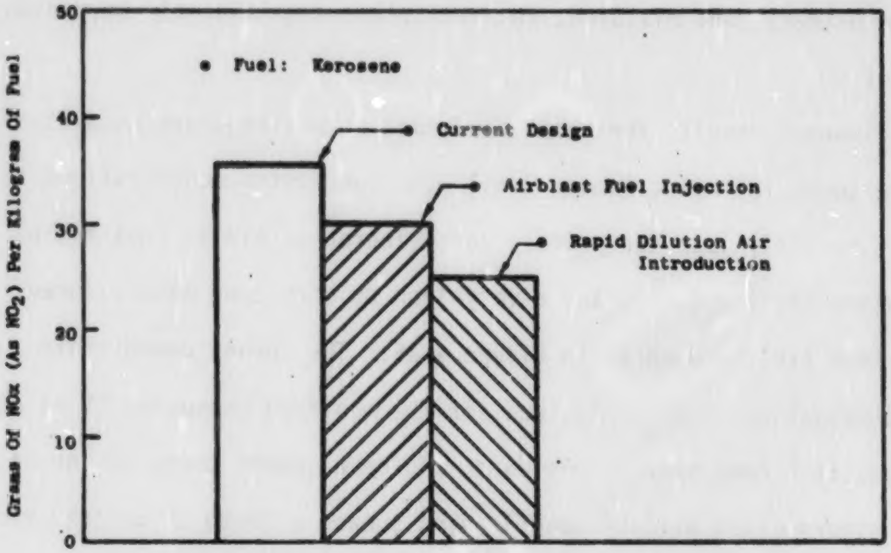


Figure 4.23[95] Reduction In the Level of NO_x Emissions at Take-off Power from an Aircraft Gas Turbine Using Combustor Design Modifications

primary zone combustion at stoichiometric conditions. Figure 4.23 also shows that rapid introduction of dilution air to reduce the dwell time of the combustion gases at high temperatures resulted in a 30 percent reduction in NO_x emissions.

As discussed in Chapter 3, reduced levels of NO_x emissions may be obtained by operating with much leaner or much richer average primary zone fuel to air ratio than stoichiometric. Figure 4.24 shows this effect and shows that the use of lean primary zone mixtures, in particular, significantly decreases emissions of NO_x.

Experimental results for tests performed at Westinghouse on a standard gas turbine combustor using Number 2 oil with successive modifications to reduce residence time (II), lean the primary zone (increased air to fuel ratio) with reduced residence time (III), and a very lean primary zone with reduced residence time (IV), are shown in Figure 4.25. The curves demonstrate about a 40 percent reduction in NO_x emissions for the modified combustor IV as compared to the production combustor I. Research and development tests on one-half scale combustors using primary zone leaning showed a 10 to 20 percent reduction in NO_x for oil and gas fuels, respectively.[98]

Results of research and development tests performed by Westinghouse on a one-half scale combustor without and with simulated exhaust gas recirculation (EGR) are shown in Figures 4.26 and 4.27 when burning Number 2 oil and natural gas, respectively. The figures show that a 38 percent and 30 percent reduction in NO_x emissions was accomplished when using EGR for oil and gas combustion, respectively. This was accomplished at an EGR rate of 26 percent and the effect on the HC and CO emissions was beneficial.[100]

Solar, under contract to NASA-Lewis Research Center, conducted rig tests of "can" type combustors to demonstrate practical combustor technology for the

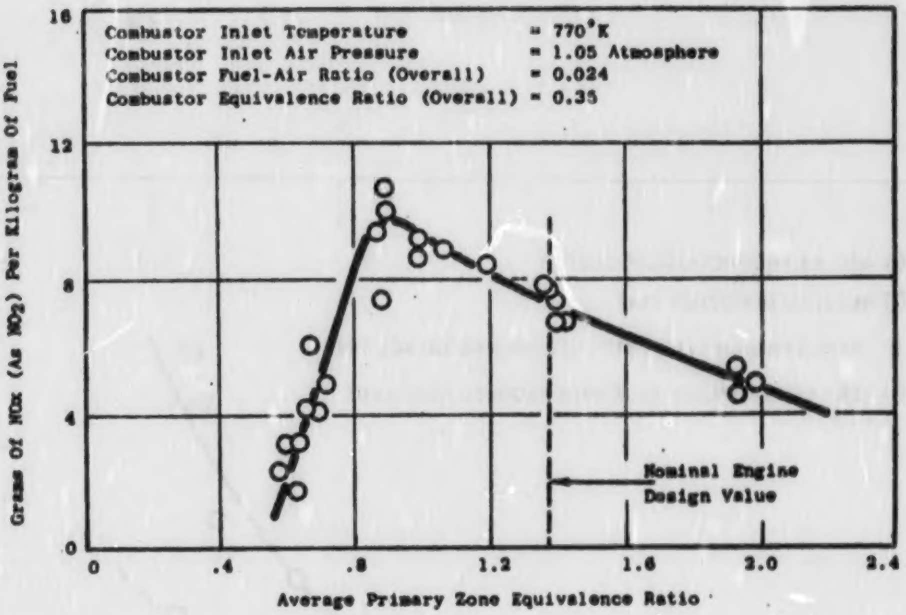


Figure 4.24 [96] Effect of Primary Zone Equivalence Ratio on NO_x Emissions from an Aircraft Gas Turbine Combustor at Simulated Take-off Power

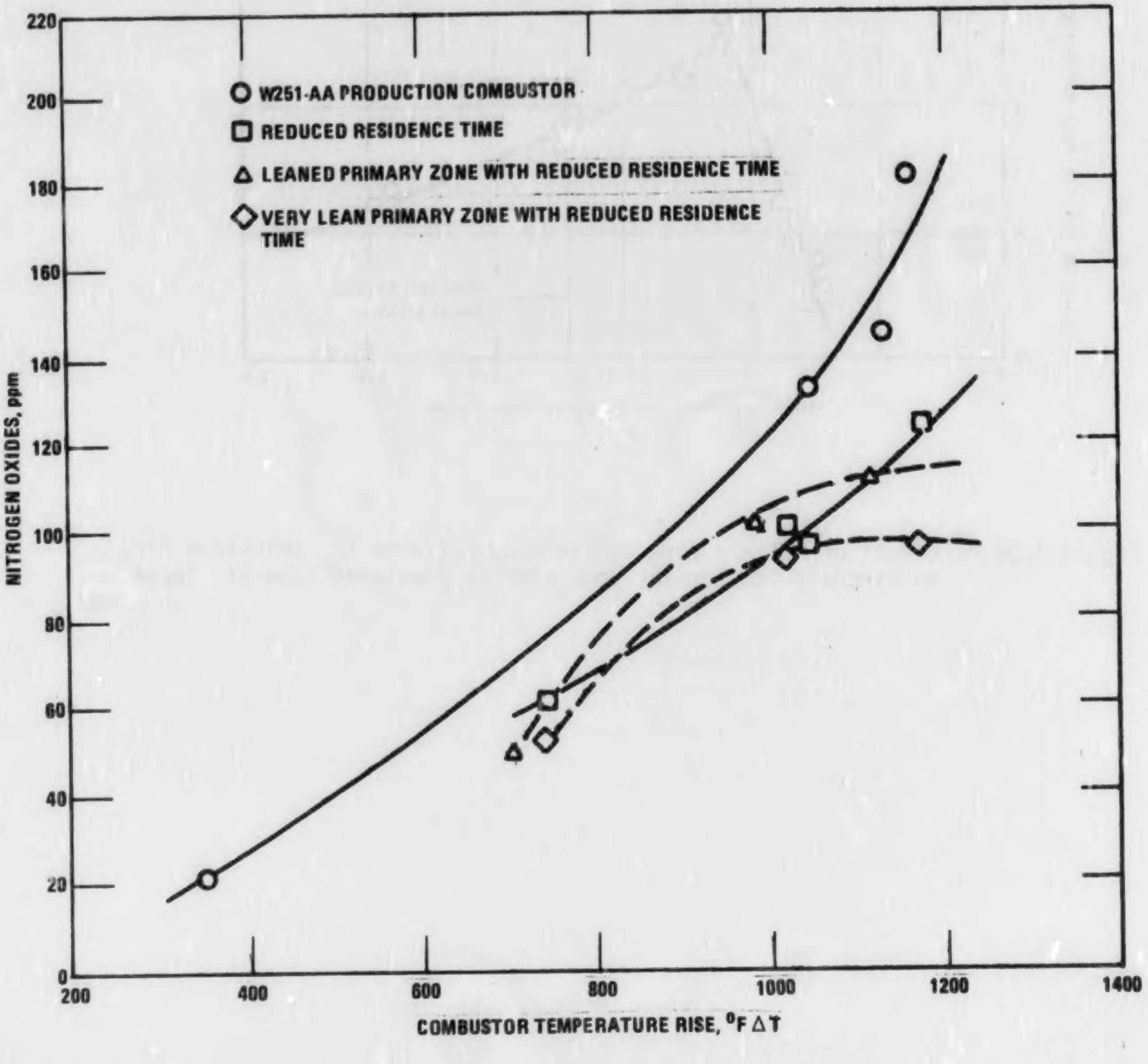


Figure 4.25[97]. NO_x emissions versus combustor temperature rise for a production combustor and several modified versions of the production combustor.

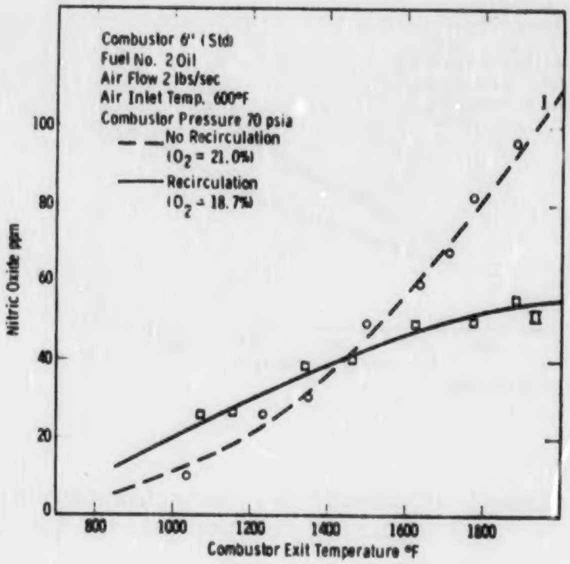


Figure 4.26[99] Effect of Exhaust Gas Recirculation on NO Emissions for a Combustor burning No. 2 oil

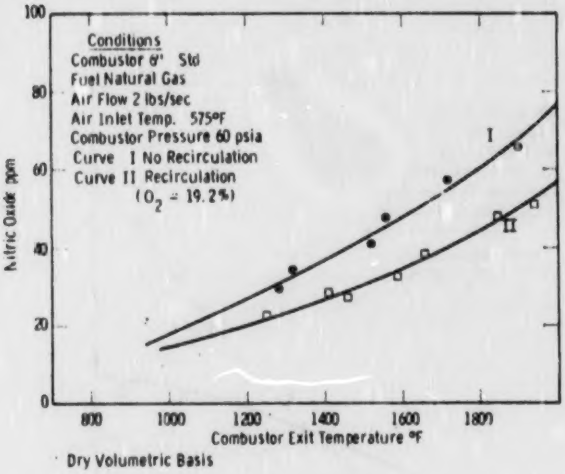


Figure 4.27[101] Effect of Exhaust Gas Recirculation on NO Emissions for a Combustor Burning Natural Gas

reduction of pollutants from future generation aircraft turbines. One goal was to reduce the emissions of NO_2 at cruise conditions from the present levels of approximately 18 to 20 gm NO_2/kg fuel (220 ppm to 247 ppm at 15 percent O_2) to a level of 5 gm NO_2/kg fuel (62 ppm at 15 percent O_2) or less.[102] [1/] During these research and development tests, two basic configurations of combustors were tested - the Vortex Airblast combustor (VAB) and the Jet Induced Circulation Combustor (JIC). Both the VAB and the JIC concepts utilized lean primary zones, prevaporization of the fuel and premixing of the air and fuel. The basic VAB combustor is illustrated in Figure 4.28.

Figures 4.29, 4.30, 4.31, 4.32, and 4.33 illustrate design modifications which were made to the basic VAB combustor during the test series to improve combustion stability, improve the fuel and air mixing and reduce emissions of NO_x, HC, and CO. The data resulting from tests of the basic and modified VAB combustors are shown in Figures 4.34A through 4.34E. The VAB combustor demonstrated a NO_x level of about 1.1 gm NO_2/kg fuel at 0 percent humidity (13.6 ppm at 15 percent O_2) with essentially 100 percent combustion efficiency at simulated cruise combustor conditions of 5 atmospheres inlet pressure and 1040°F inlet temperature when burning Jet A-1 fuel.[110] This represents about a 94 percent reduction in NO_x emissions as compared to the present generation aircraft turbine (e.g., 18 to 20 gm NO_2/kg fuel). It is interesting to note that the NO_x emissions from the VAB combustor did not show dependency on inlet pressure, as does a conventional combustor.[111] Figure 4.34B shows the effects of fuel on NO_x emissions, and Figures 4.34D and E show the effects of combustor inlet temperature on NO_x and CO emissions, respectively.

The basic JIC combustor is shown in Figure 4.35 with fuel injection and combustor design modifications shown in Figure 4.36. Data results are presented in Figures 4.37A through E. As can be observed, the VAB combustor concept

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[1/] A conversion: ppm @ 15% O_2 = 12.34 X gm NO_2/kg fuel was used for these figures.

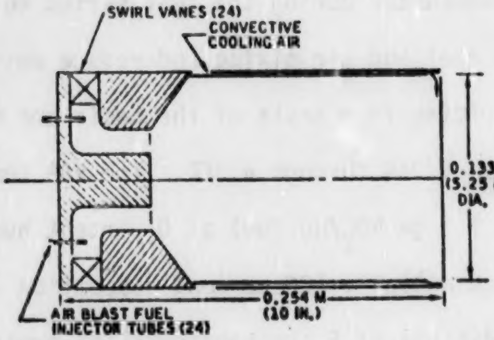

SWIRL VANES (24)
CONVECTIVE COOLING AIR
0.133 M (5.25 IN.) DIA.
0.254 M (10 IN.)
AIR BLAST FUEL INJECTOR TUBES (24)

Figure 4.28[103] Details of the Basic Vortex Airblast (VAB) Combustor

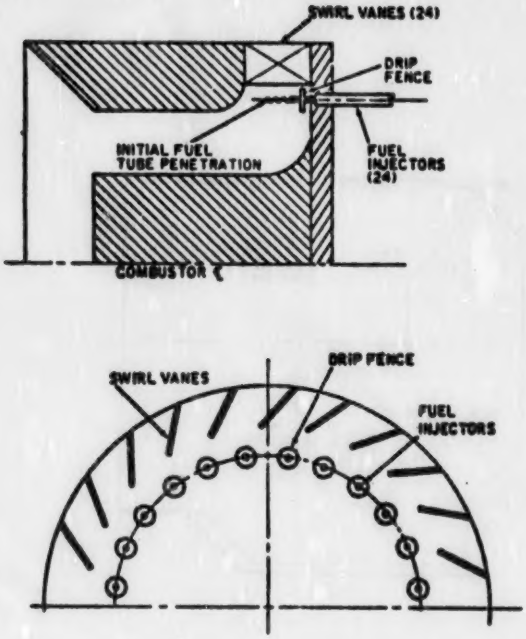

SWIRL VANES (24)
DRIP FENCE
INITIAL FUEL TUBE PENETRATION
FUEL INJECTORS (24)
COMBUSTOR ℄
SWIRL VANES
DRIP FENCE
FUEL INJECTORS

Figure 4.29[104] VAB Combustor - Fuel Injection Modification

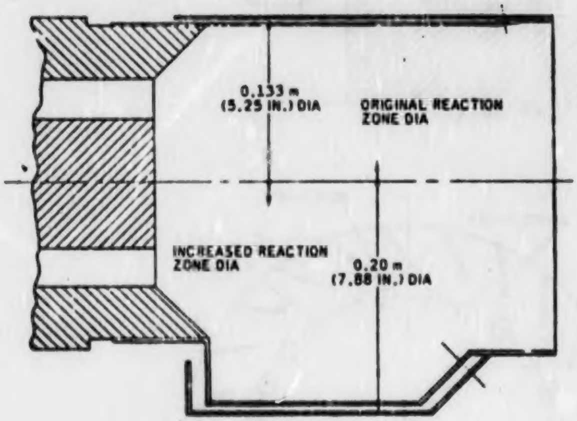

0.133 m
(5.25 IN.) DIA
ORIGINAL REACTION
ZONE DIA
INCREASED REACTION
ZONE DIA
0.20 m
(7.88 IN.) DIA

Figure 4.30[105] VAB Combustor - Reaction Zone Diameter Increase

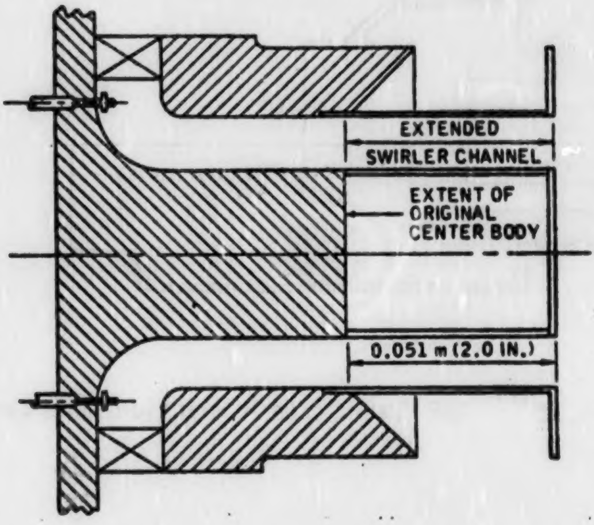

EXTENDED
SWIRLER CHANNEL
EXTENT OF
ORIGINAL
CENTER BODY
0.051 m (2.0 IN.)

Figure 4.31[106] VAB Combustor - Swirler Throat Length Increase

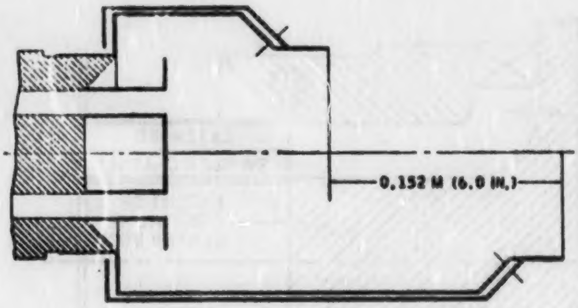


Figure 4.32[107] VAB Combustion - Reaction Zone Length Increase

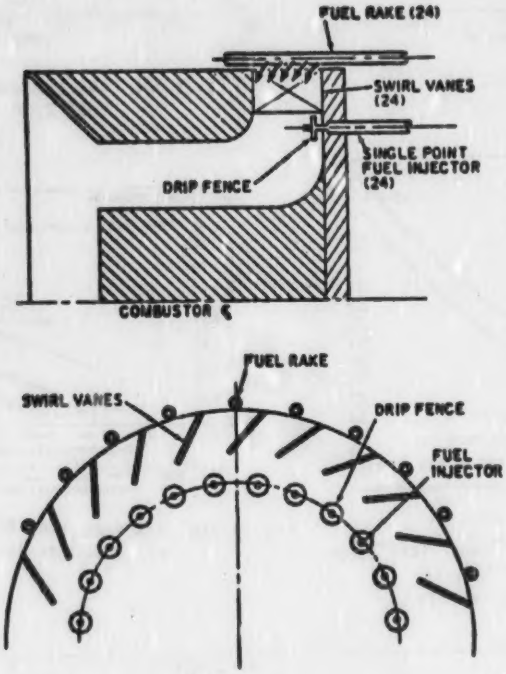

FUEL RAKE (24)
SWIRL VANES (24)
SINGLE POINT FUEL INJECTOR (24)
DRIP FENCE
COMBUSTOR ℄
FUEL RAKE
SWIRL VANES
DRIP FENCE
FUEL INJECTOR

Figure 4.33 [108] VAB Fuel Injection Modification

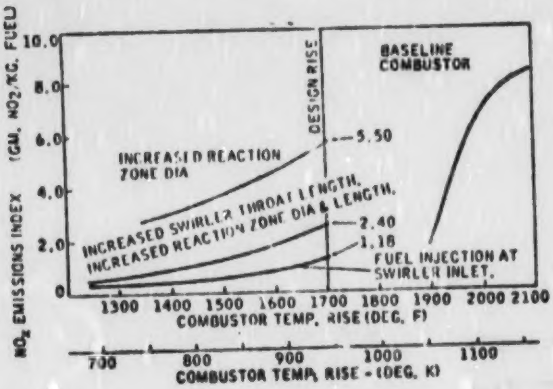


Fig. 4.34A VAB combustor NO_x test results;

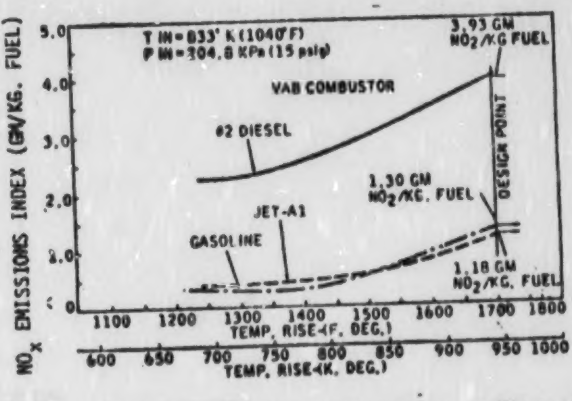


Fig. 4.34B VAB combustor test results - effect of fuel type

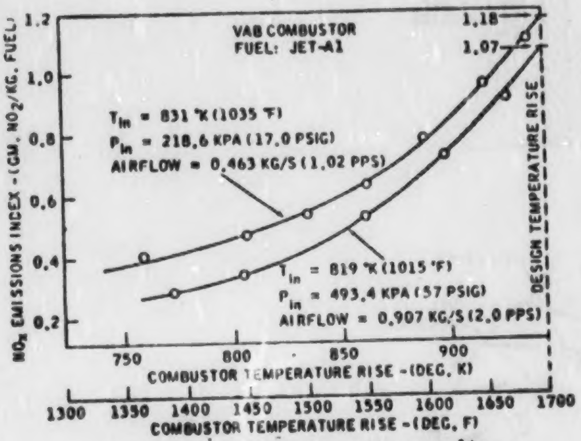


Fig. 4.34C VAB combustor test results;

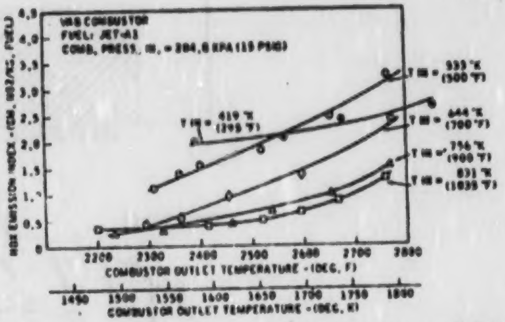


Fig. 4.34D VAB combustor NO_x test results - effect of combustor inlet temperature

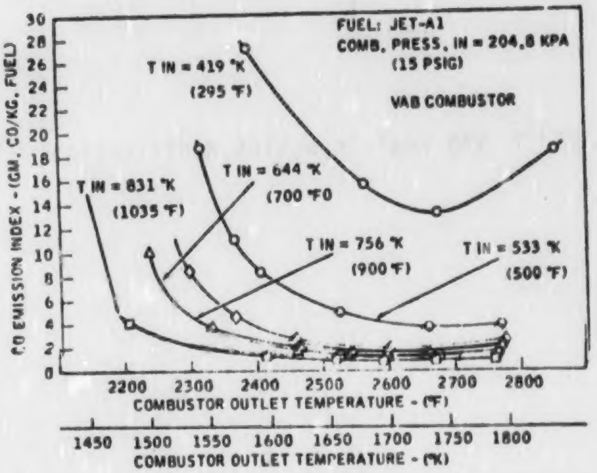


Fig. 4.34E VAB combustor CO test results - effect of combustor inlet temperature

Figures 4.34[109] A, B, C, D, and E Results from Rig Tests of the VAB Combustor

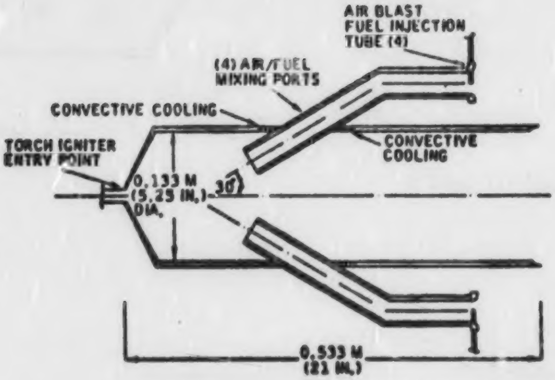

AIR BLAST
FUEL INJECTION
TUBE (4)
(4) AIR/FUEL
MIXING PORTS
CONVECTIVE COOLING
TORCH IGNITER
ENTRY POINT
CONVECTIVE
COOLING
0.133 M
(5.25 IN.)
DIA.
30°
0.533 M
(21 IN.)

Figure 4.35[112] Basic Jet Induced Circulation (JIC) Combustion Details

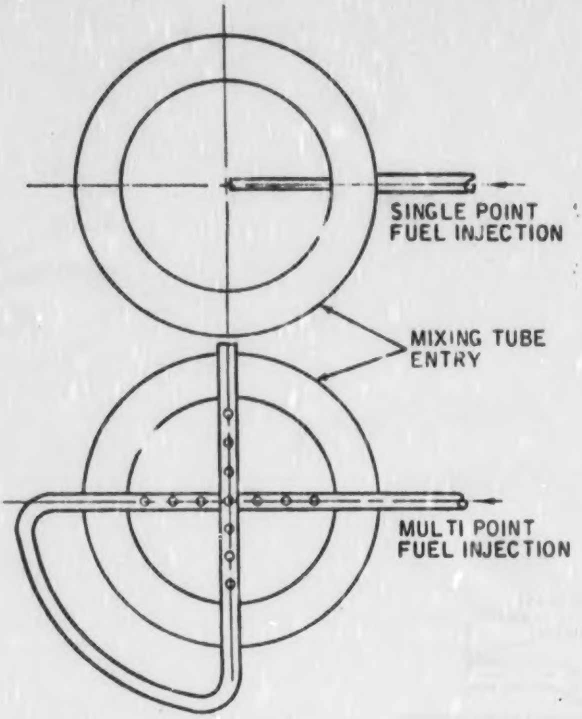

SINGLE POINT
FUEL INJECTION
MIXING TUBE
ENTRY
MULTI POINT
FUEL INJECTION

Figure A JIC Combustor Fuel Injector Modification

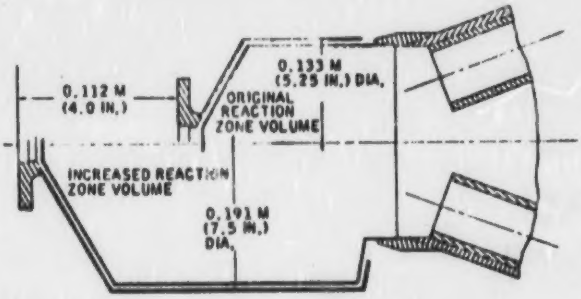

0.112 M
(4.0 IN.)
0.133 M
(5.25 IN.) DIA.
ORIGINAL
REACTION
ZONE VOLUME
INCREASED REACTION
ZONE VOLUME
0.191 M
(7.5 IN.)
DIA.

Figure B JIC Combustor Reaction Zone Volume Increase

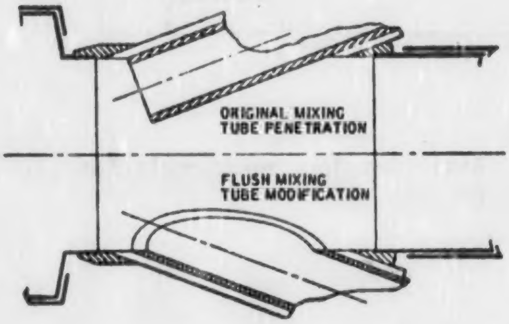

ORIGINAL MIXING
TUBE PENETRATION
FLUSH MIXING
TUBE MODIFICATION

Figure C JIC Combustor Mixing Tube Modifications

Figure 4.36[113] JIC Combustor Modifications

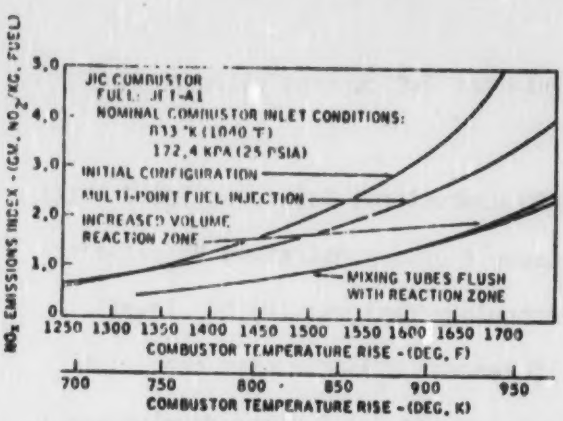


Figure A JIC combustor NO_x test results

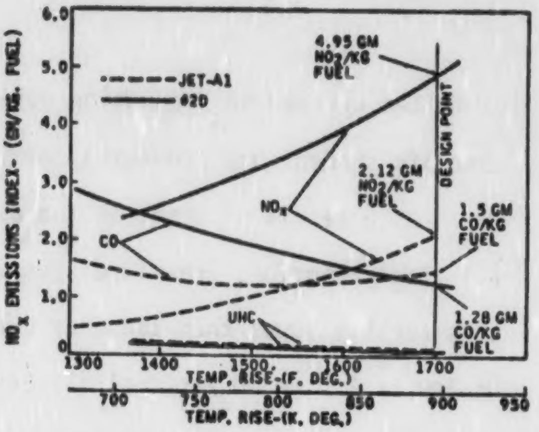


Figure B JIC combustor test results -- effect of fuel type

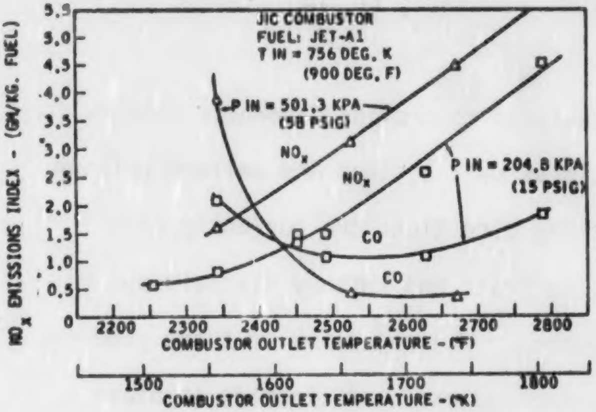


Figure C JIC combustor test results -- effect of combustor inlet pressure

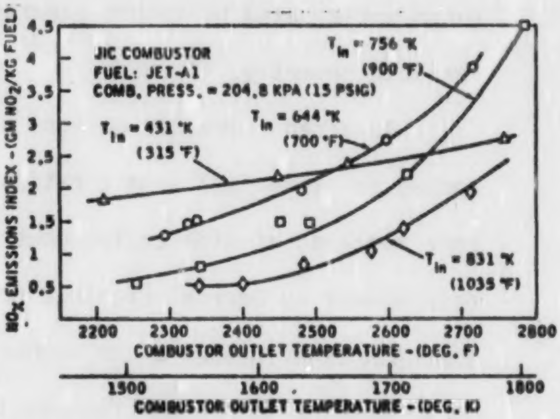


Figure D JIC combustor NO_x test results-- effect of combustor inlet temperature

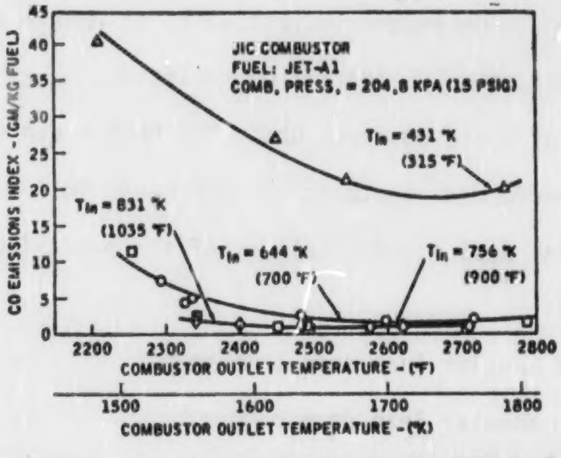


Figure E JIC combustor CO test results--effect of combustor inlet temperature

Figure 4.37[114] A, B, C, D, E

Results from Rig Tests of the JIC Combustor exhibited greater NO_2 reduction potential than the JIC concept although both concepts showed very promising results.

As a result of tests on the JIC and VAB combustors, Solar concluded that: (1) lean reaction, pre-mixed systems are capable of demonstrating NO_x levels considerably lower than those of current technology designs; (2) NO_x levels as low as 1.0 gm NO_2/kg fuel (12.34 ppm @ 15 percent O_2) at cruise condition were demonstrated with very low HC and CO levels; and (3) further development of these concepts is necessary to extend combustor operation over the full load range anticipated by future aircraft engines; possibly by fuel staging or variable geometry.[115]

Advanced combustor designs using features to provide favorable primary combustion zone fuel to air ratios at idle as well as lean and uniform primary zone mixtures at high engine power operating conditions are currently under development by General Electric Company and Pratt and Whitney Aircraft under the NASA experimental clean combustor program. The objective of these programs is to develop technology for use in existing and advanced aircraft turbines to meet the 1979 standards for emissions of CO, HC, NO_x, and smoke as defined by EPA for aircraft. The program objective is to meet the 1979 standards without the use of water injection to control NO_x emissions.

For Phase I of their contract under the NASA clean combustor program, General Electric performed rig tests of four basic full scale combustor design concepts and several design modifications of the basic four. The four configurations are:

1) A single annular lean dome combustor
2) A double annular lean dome combustor
3) A radial/axial staged combustor
4) The NASA swirl-can combustor

In addition, the standard CF6-50 combustor was tested to obtain baseline data. This basic combustor configuration is shown in Figures 4.38 and 4.39.

The lean dome single annular combustor is shown in Figure 4.40. The lean dome single annular configuration is basically a CF6-50 combustor modified to increase the air flow through the primary reaction zone, thereby decreasing the primary zone equivalence ratio. Variable geometry would be needed with this configuration to provide stable combustion throughout the operating load range.

Schematics of the lean dome double annular combustor are shown in Figures 4.41 and 4.42. In this configuration, both annuli are fueled at high power conditions with only the outer annulus fueled at low power conditions. This two stage combustion could, therefore, eliminate the need for variable geometry as required for the lean dome single annular configuration. Again, the primary zone air flow of the double annular combustor was increased to reduce the equivalence ratio.

The radial/axial staged combustor configuration is shown in Figures 4.43 and 4.44. Both radial and axial fuel staging provisions are incorporated. The pilot stage (as shown in Figure 4.43) is of conventional design and was sized for low power operation. Both stages operate at high power conditions and some degree of fuel and air premixing occurs in the main stage combustor. Lean combustion in the primary zone was also a design objective in this configuration.

The lean dome double annular combustor and the radial/axial combustor both used airblast fuel injection wherein a portion of the compressor discharge air is utilized to atomize the fuel and provide better fuel and air mixing. The use of airblast fuel injection results in reduced NO_x emissions when compared to a conventional high pressure fuel nozzle as shown in Figure 4.23.

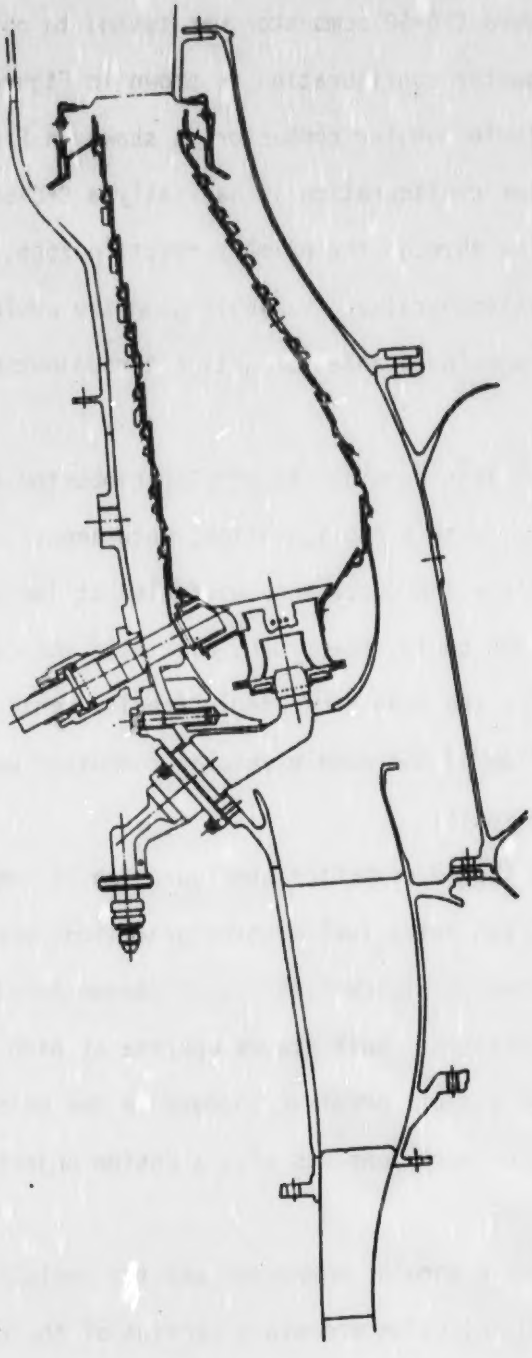


Figure 4.38[116] Production CF6-50 Engine Combustor

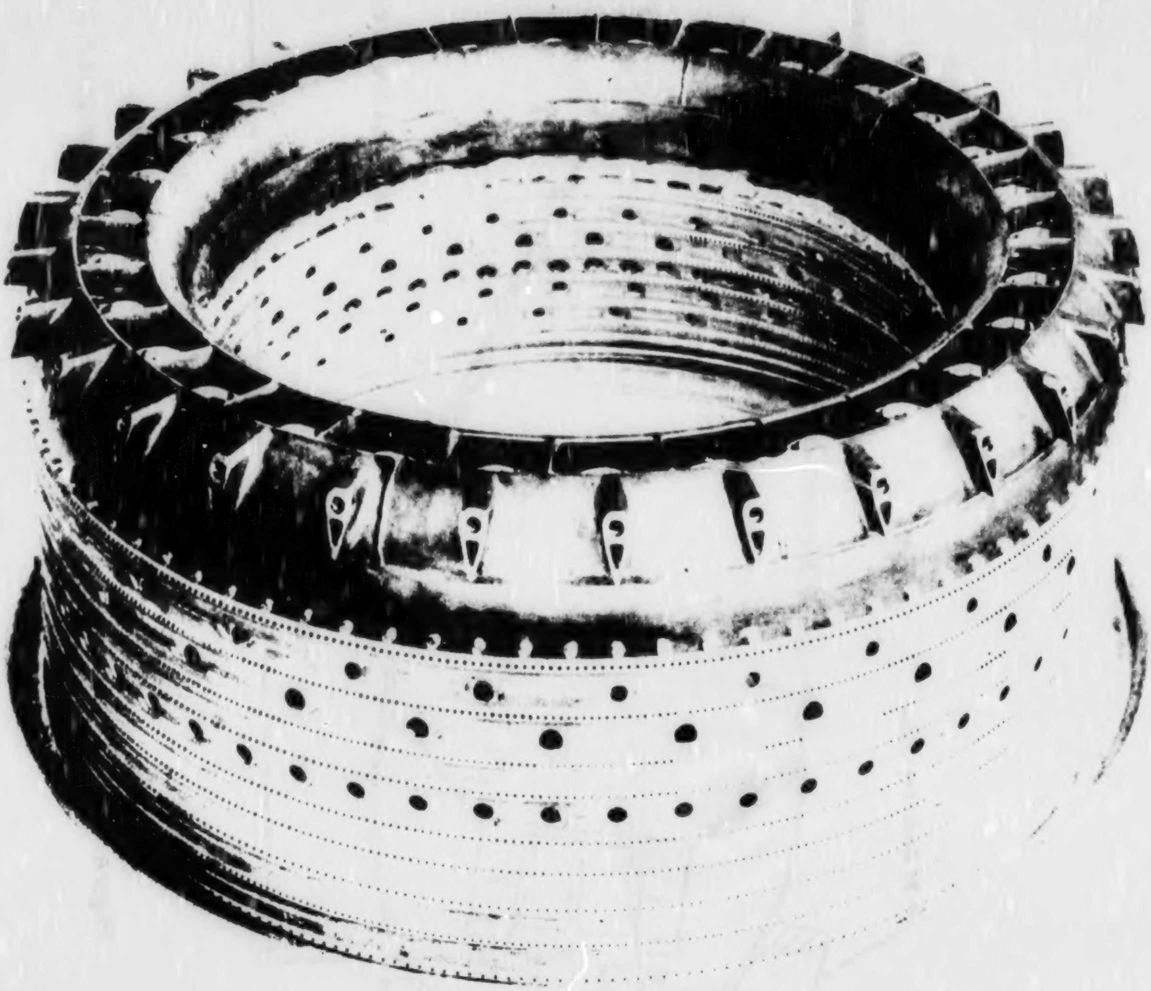


Figure 4.39[117]. Production CF6-50 combustor assembly.

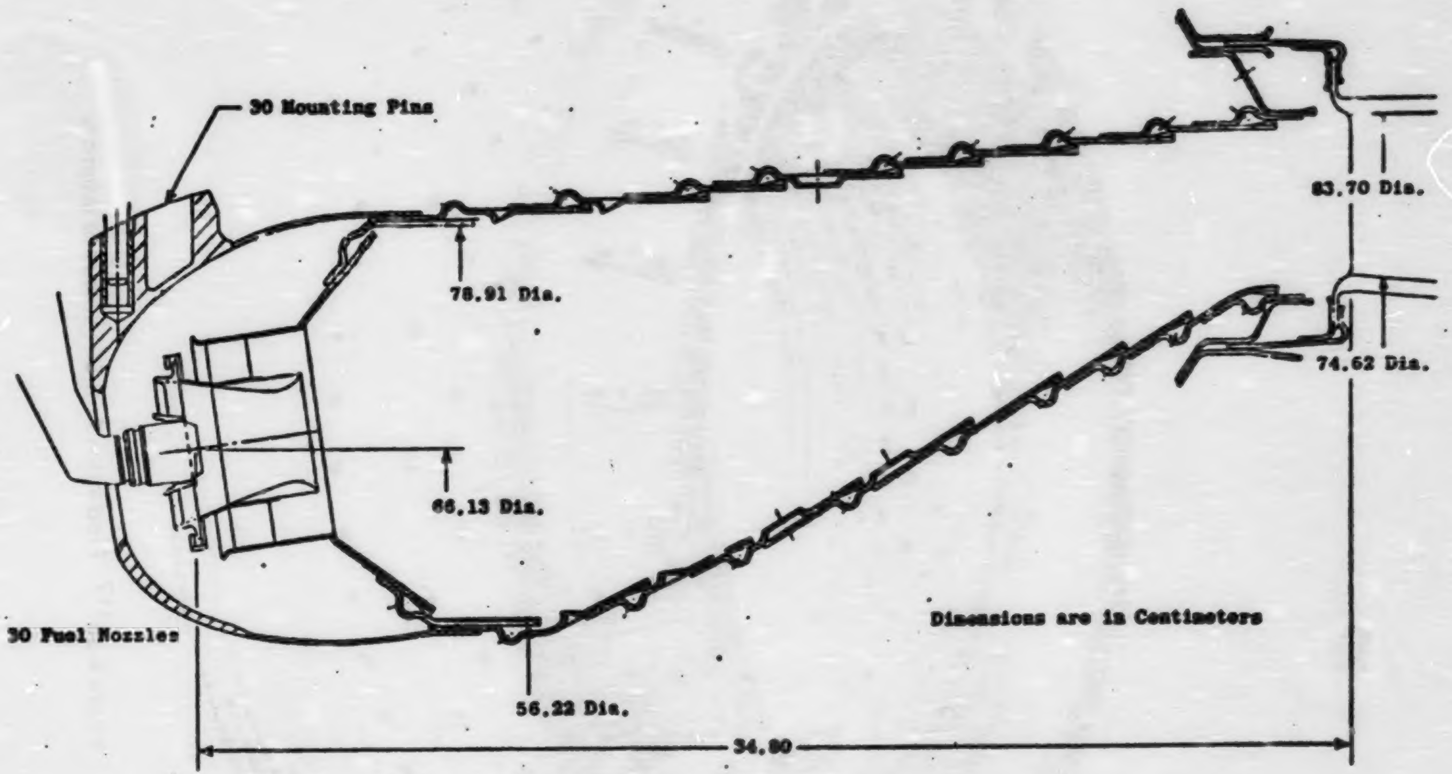

30 Mounting Pins
83.70 Dia.
78.91 Dia.
74.62 Dia.
66.13 Dia.
30 Fuel Nozzles
Dimensions are in Centimeters
56.22 Dia.
34.80

Figure 4.40[118] General Arrangement, Single Annular Combustor.

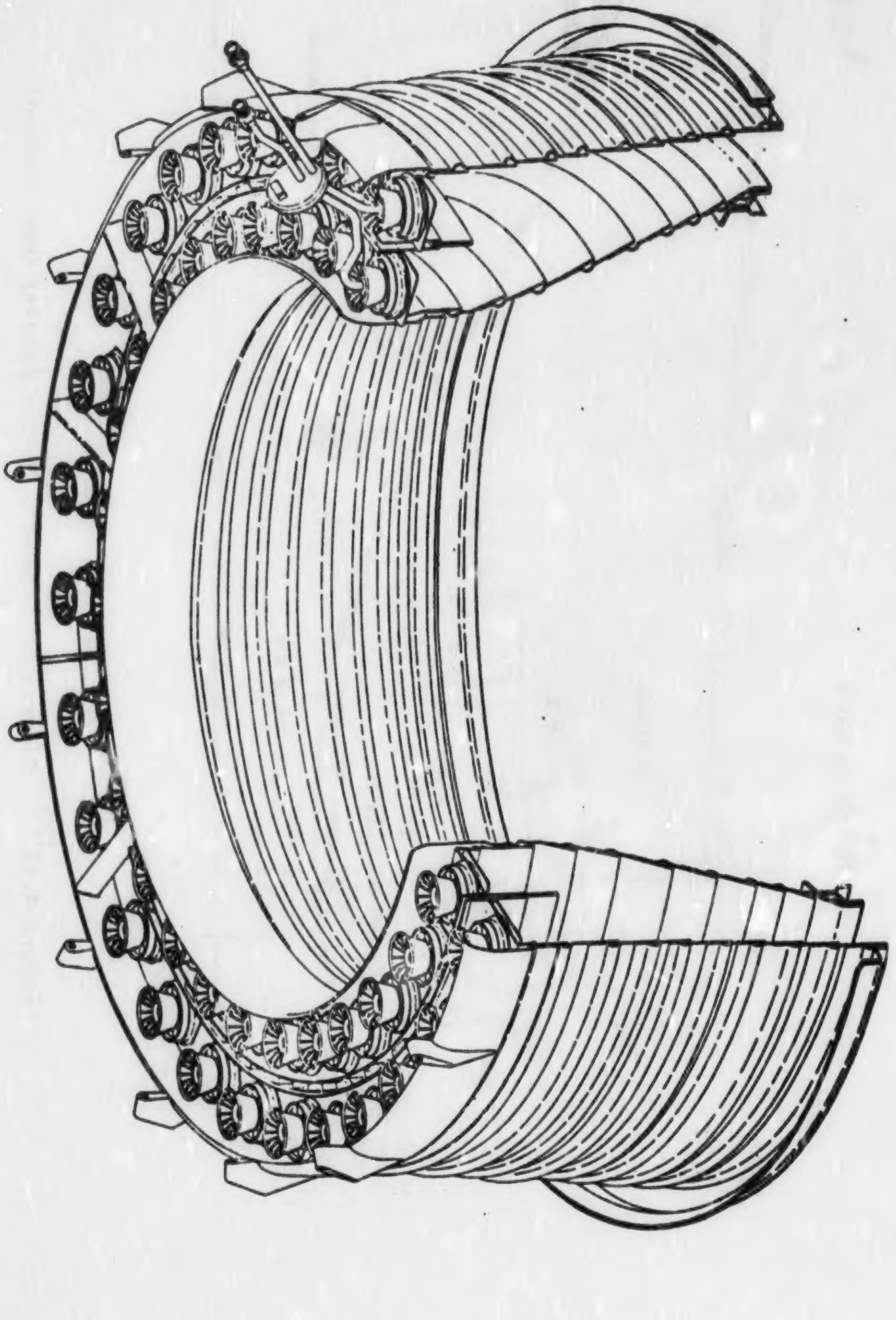


Figure 4.41[119] Lean Dome Double Annular Combustor for CF6-50 Engine

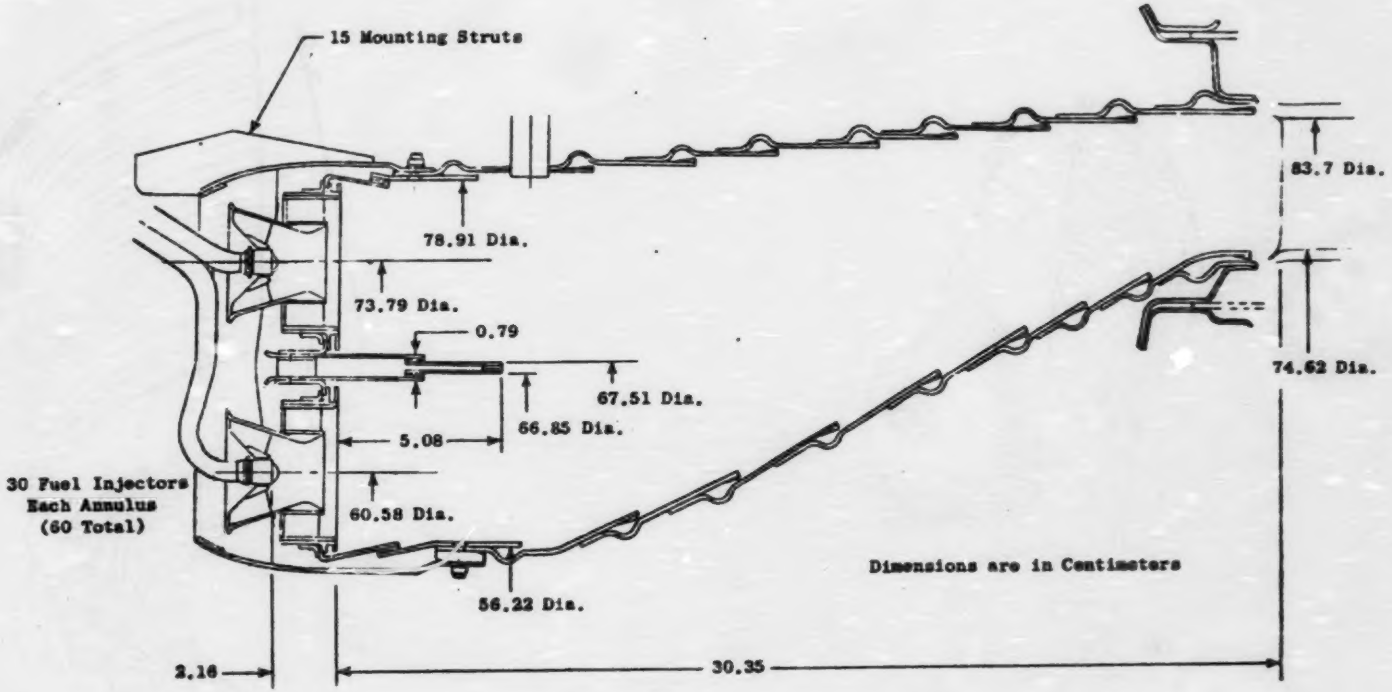

15 Mounting Struts
83.7 Dia.
78.91 Dia.
73.79 Dia.
0.79
67.51 Dia.
66.85 Dia.
5.08
74.62 Dia.
30 Fuel Injectors
Each Annulus
(60 Total)
60.58 Dia.
Dimensions are in Centimeters
56.22 Dia.
2.16
30.35

Figure 4.42[120] General Arrangement, Double Annular Dome Combustor.

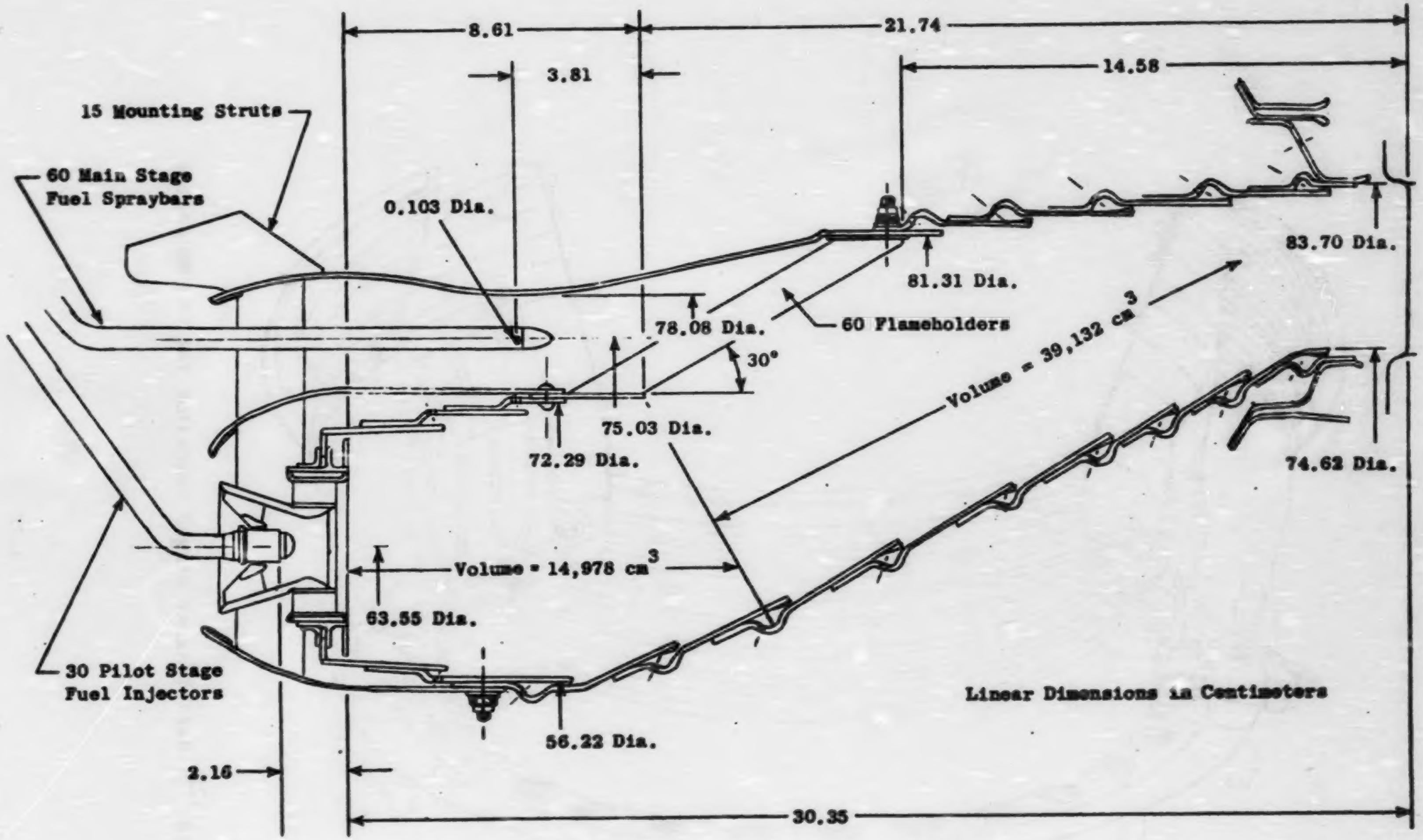

8.61
21.74
3.81
14.58
15 Mounting Struts
60 Main Stage Fuel Spraybars
0.103 Dia.
81.31 Dia.
83.70 Dia.
78.08 Dia.
60 Flameholders
30°
Volume = 39,132 cm³
75.03 Dia.
72.29 Dia.
74.62 Dia.
Volume = 14,978 cm³
63.55 Dia.
30 Pilot Stage Fuel Injectors
Linear Dimensions in Centimeters
56.22 Dia.
2.16
30.35

Figure 4.43 [121] General Arrangement, Radial/Axial Staged Combustor

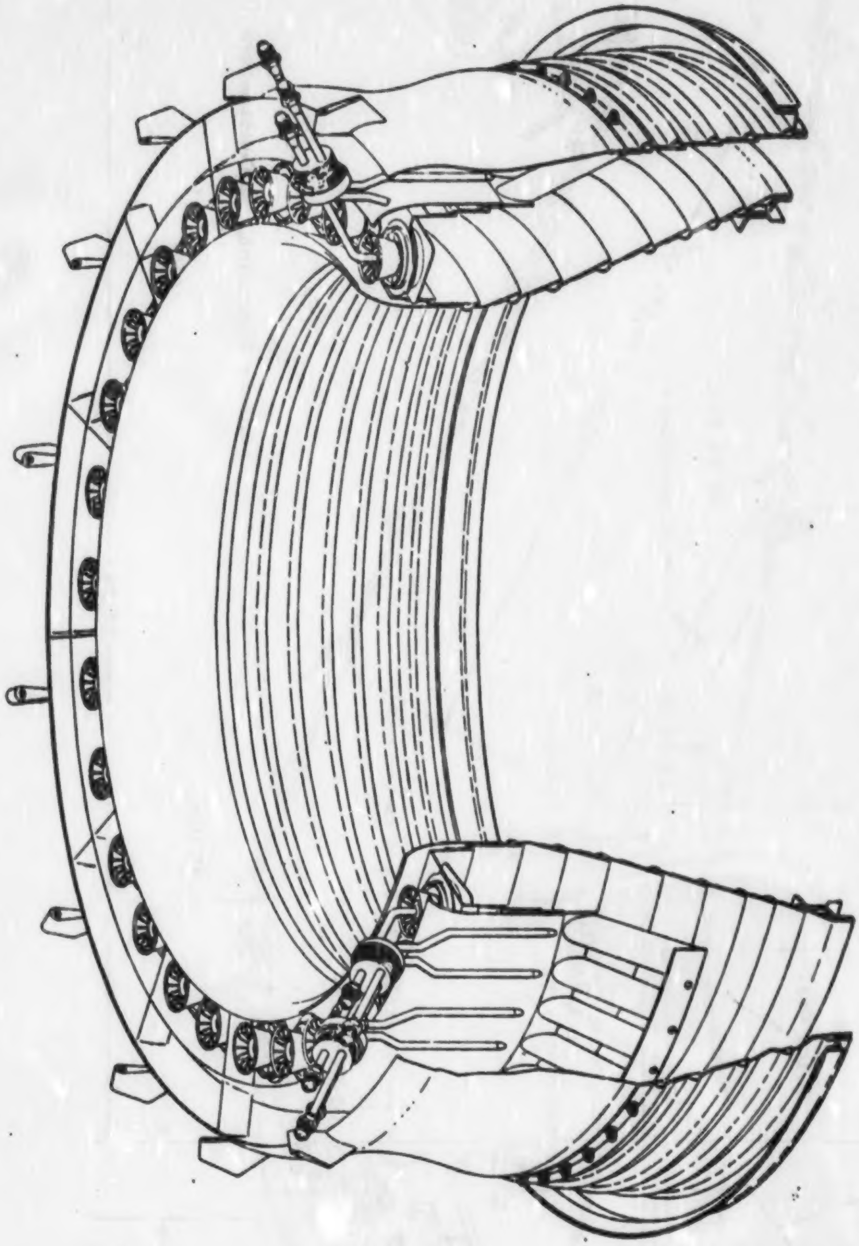


Figure 4.44 [122] **Radial/Axial Staged Combustor for CF6-50 Engine.**

A schematic of the baseline swirl-can combustor design is shown in Figure 4.45. This concept consists of a modular array of carburating swirl cans, each with an axial air swirler and a stabilizing plate. Each module is designed to premix the fuel with the air in the carburetor, swirl the fuel-air mixture, stabilize combustion in the swirl can wake and provide mixing areas between the bypass air through the swirl can array and the hot gases in the wake of the swirl can modules. With this configuration, fuel staging was investigated by controlling fuel flows to the various modules.

NO_x emissions versus fuel-to-air ratio for the CF6-50 production combustor and the best configuration of each major design approach (discussed above) are shown in Figure 4.46. The double annular and radial/axial configurations obviously resulted in the greatest reduction of NO_x emissions. A summary of emissions from the double annular and radial/axial "best configurations" is shown in Table 4.6. The table shows that NO_x reductions, as compared to the production CF6-50 turbine, at standard day take-off conditions, were 51.4 and 60 percent for the double annular and radial/axial configurations, respectively. The reductions in CO and HC emissions at idle conditions are also shown in Table 4.6.

For Phase I of their contract under the NASA clean combustor program, Pratt and Whitney Aircraft Division performed rig tests of three basic full scale combustor design concepts and several design modifications of the basic three. The three configurations are:

(1) Swirl-Can combustor concept

(2) Staged premix combustor concept

(3) Swirl Vorbix combustor concept

Baseline data from earlier testing of the production JT9D-7 annular combustor were used for comparison purposes.

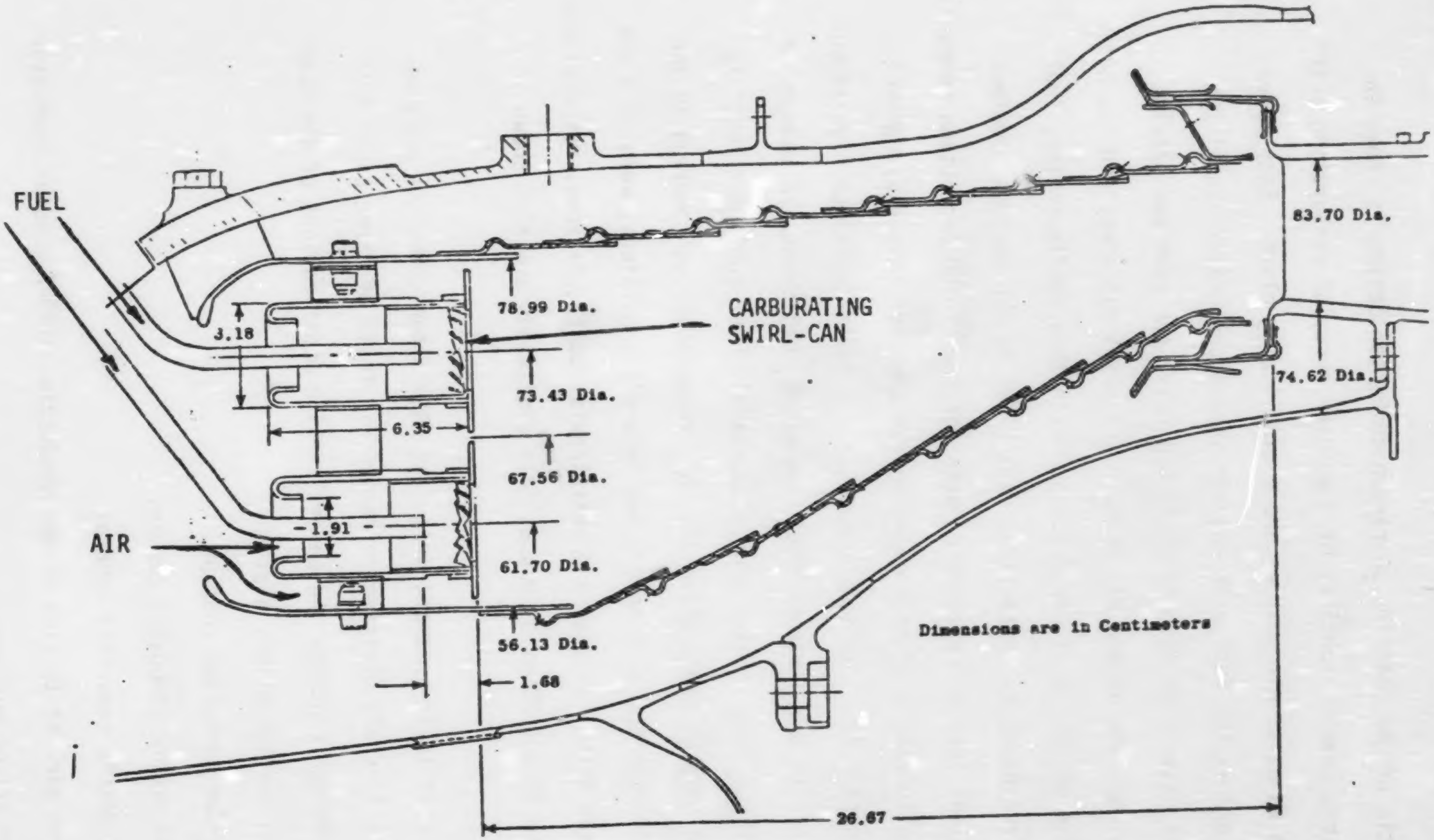

FUEL
AIR
CARBURATING SWIRL-CAN
3.18
6.35
1.91
1.68
78.99 Dia.
73.43 Dia.
67.56 Dia.
61.70 Dia.
56.13 Dia.
83.70 Dia.
74.62 Dia.
26.67
Dimensions are in Centimeters

Figure 4.45[123] Baseline Swirl-Can-Modular Combustor Design

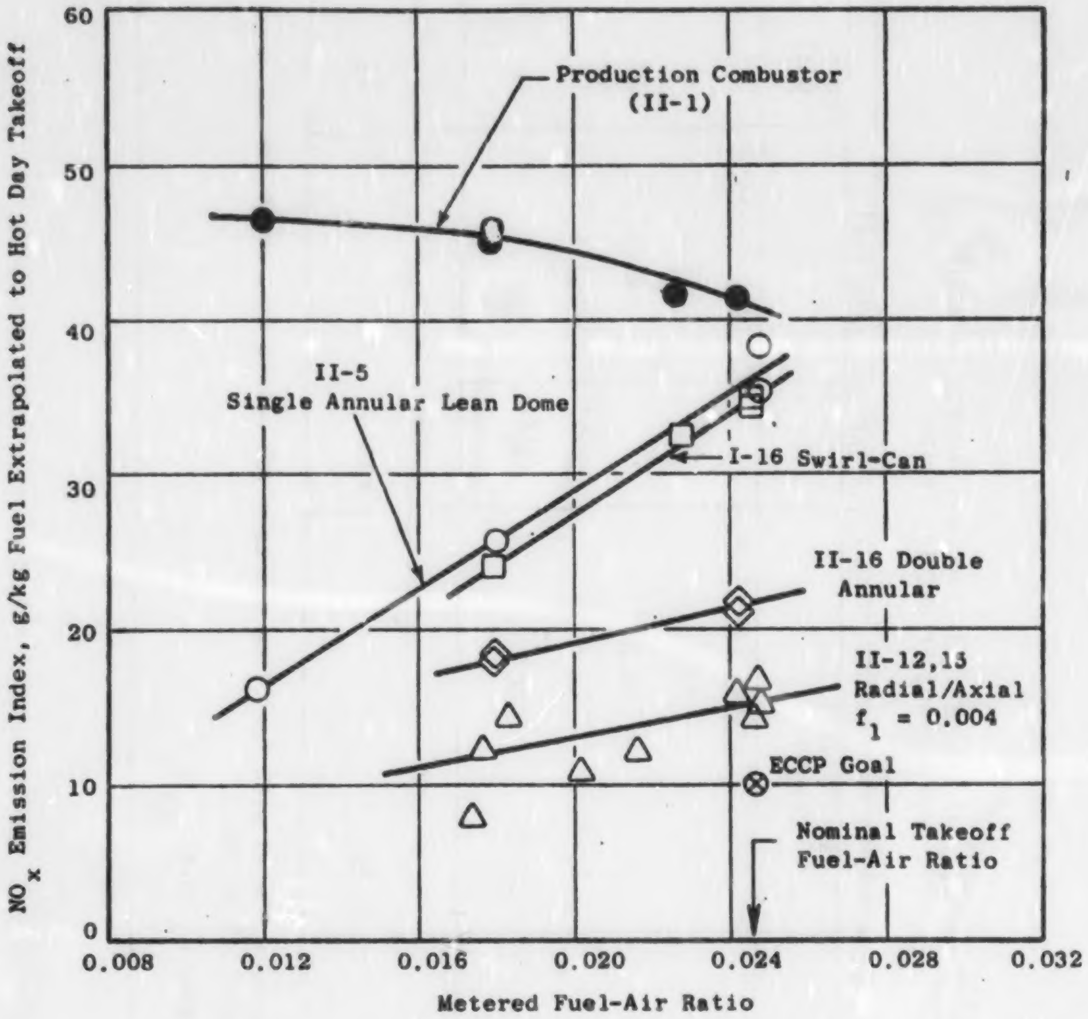


Figure 4.46[124] NO_x Emissions Levels, Best Configuration of Each Major Design Approach

NOTE: At a fuel-to-air ratio of 0.20, NO_x @ 15% $O_2 \approx$ 12.3 (Emission Index)

Table 4.6[125] Summary of Exhaust Emissions and Percent Reduction from Current Production Engine

	Emission Indices, g/kg Fuel					
	Radial/Axial Staged		Double Annular		Program Goal	Current Production Engine
		Percent Reduction		Percent Reduction		
HC at Standard Day Idle	2	92.3	6	77.8	4	27
CO at Standard Day Idle	28	58.2	34	49.3	20	67
NO_x at Hot Day Takeoff	~ 14$^{(1)}$	68.2	21$^{(2)}$	52.3	10	44
NO_x at Standard Day Takeoff	~14$^{(1)}$	60	17$^{(2)}$	51.4	--	35
	← Smoke Index, SAE, SN →					
Peak Smoke	< 15		<15		15	12

(1). Extrapolated to true engine P_3 level with fuel flow splits which result in a projected combustion efficiency level of 99.85%

(2) Extrapolated to true engine P_3 level

All tests were performed using Jet-A fuel. Figure 4.47 is a schematic of the swirl can combustor concept. This combustor consists of a three row array simulating 120 swirl modules for the entire annulus. Only the outer module row is fueled for low power operation. This combustor configuration is similar to that described earlier for the GE swirl can combustor.

A schematic of the staged premix combustor concept is shown in Figure 4.48. This combustor consists of a primary or low power burner and a secondary high power burner with each burner having its own fuel injection system and premixing passage. Both burners operate during the high power conditions. Since HC and CO are the main pollutants of concern at low power, the pilot burner was designed to operate at an equivalence ratio of 1.0 during low power operation. At high power operation, both burners were designed to operate at an equivalence ratio of approximately 0.7.

Figure 4.49 shows a schematic of the swirl vorbix combustor concept. This combustor employs two burning zones. The pilot combustion zone only is fueled during low power conditions with both burners used for high power conditions. Fuel for the main combustion chamber is introduced and vaporized at the exit of the pilot zone. The objective of the swirl vorbix combustor is to provide a relatively long combustion residence time at low power settings to minimize CO and HC emissions and to provide rapid burning and quenching of the combustion reaction at high power levels to minimize NO_x formation. The combustor features a combination of vortex burning and mixing using swirlers as shown in Figure 4.49A and B.

Table 4.7 summarizes the test results for the best configurations of each combustor concept at sea-level take-off operations. NO_x reductions ranged from 34.6 to 60.6 percent when compared to the NO_x emissions from the production

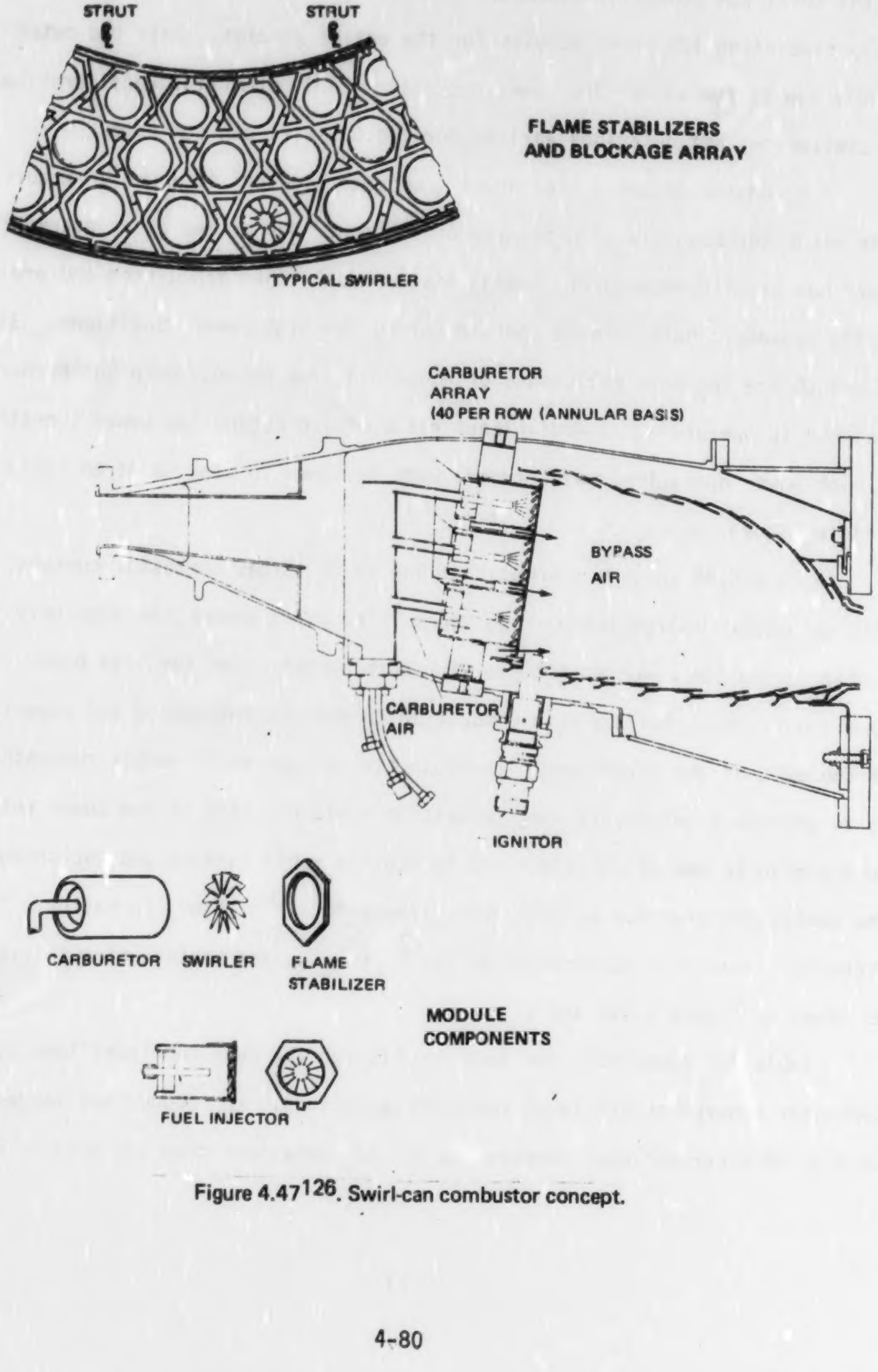
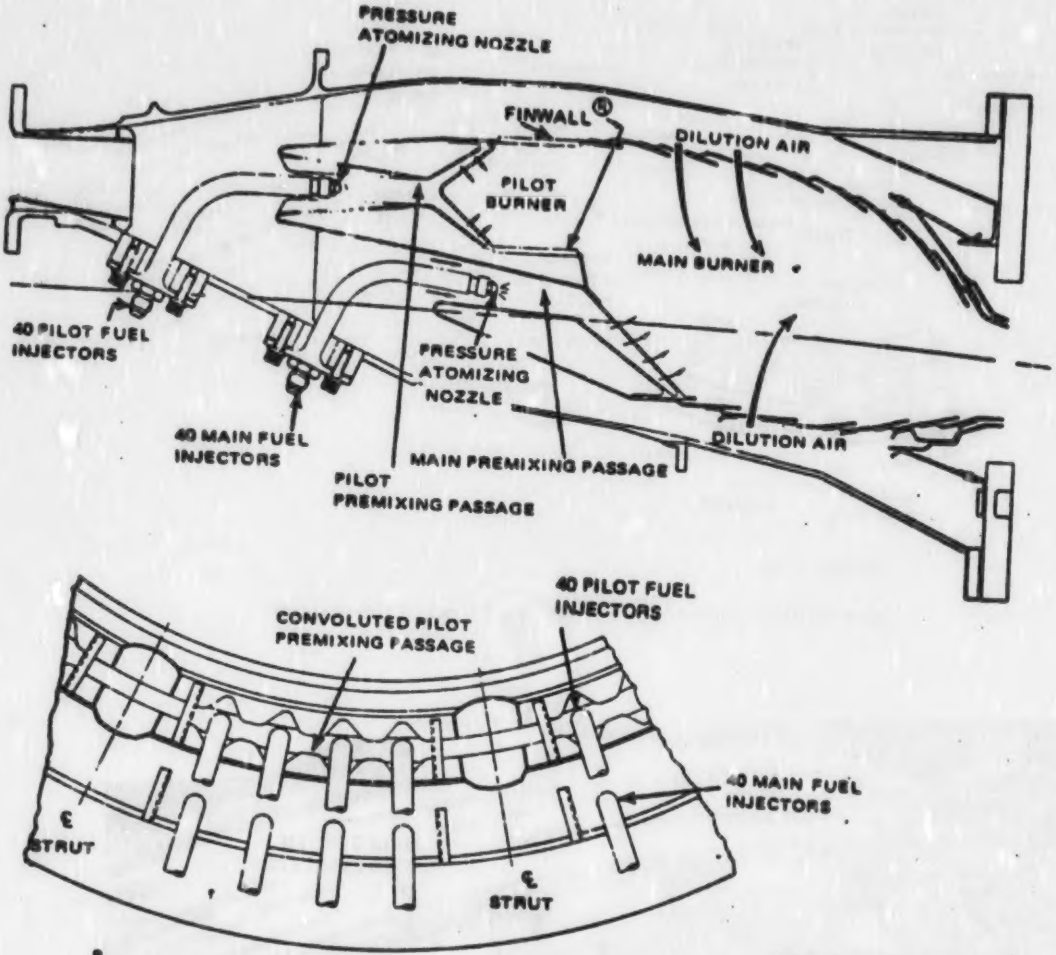
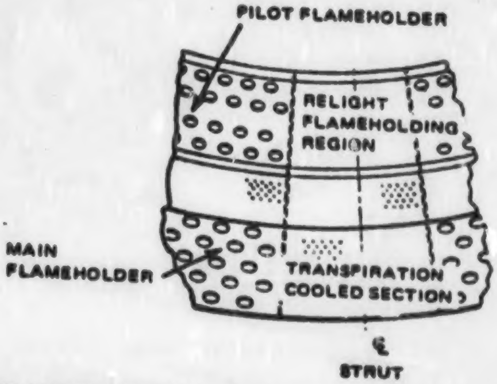

STRUT
STRUT
FLAME STABILIZERS
AND BLOCKAGE ARRAY
TYPICAL SWIRLER
CARBURETOR
ARRAY
(40 PER ROW (ANNULAR BASIS)
BYPASS
AIR
CARBURETOR
AIR
IGNITOR
CARBURETOR
SWIRLER
FLAME
STABILIZER
MODULE
COMPONENTS
FUEL INJECTOR

Figure 4.47[126]. Swirl-can combustor concept.


PRESSURE ATOMIZING NOZZLE
FINWALL®
DILUTION AIR
PILOT BURNER
MAIN BURNER
40 PILOT FUEL INJECTORS
PRESSURE ATOMIZING NOZZLE
40 MAIN FUEL INJECTORS
MAIN PREMIXING PASSAGE
DILUTION AIR
PILOT PREMIXING PASSAGE
40 PILOT FUEL INJECTORS
CONVOLUTED PILOT PREMIXING PASSAGE
40 MAIN FUEL INJECTORS
℄ STRUT
℄ STRUT

PREMIXING PASSAGE FRONT END VIEW


PILOT FLAMEHOLDER
RELIGHT FLAMEHOLDING REGION
MAIN FLAMEHOLDER
TRANSPIRATION COOLED SECTION
℄ STRUT

PILOT AND MAIN BURNER FLAMEHOLDERS

Figure 4.48 [127] Staged Premix Combustor Concept

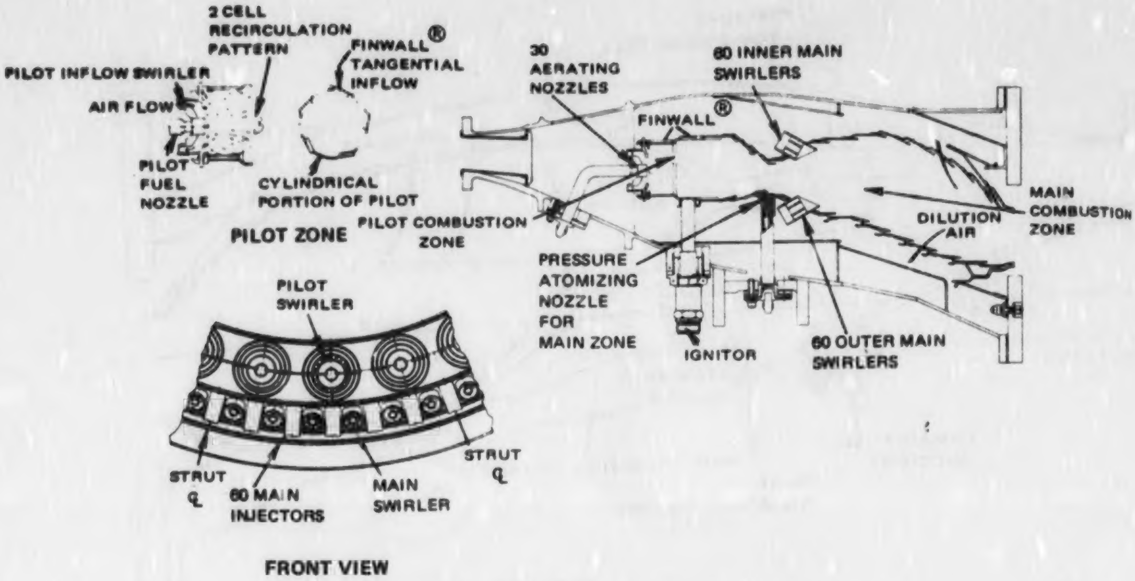

2 CELL RECIRCULATION PATTERN
FINWALL® TANGENTIAL INFLOW
PILOT INFLOW SWIRLER
AIR FLOW
PILOT FUEL NOZZLE
CYLINDRICAL PORTION OF PILOT
PILOT ZONE
30 AERATING NOZZLES
60 INNER MAIN SWIRLERS
FINWALL®
PILOT COMBUSTION ZONE
MAIN COMBUSTION ZONE
DILUTION AIR
PRESSURE ATOMIZING NOZZLE FOR MAIN ZONE
IGNITOR
60 OUTER MAIN SWIRLERS
PILOT SWIRLER
STRUT ℄
60 MAIN INJECTORS
MAIN SWIRLER
STRUT ℄
FRONT VIEW

Figure 4.49 A. Schematic of the Swirl Vorbix Combustor.

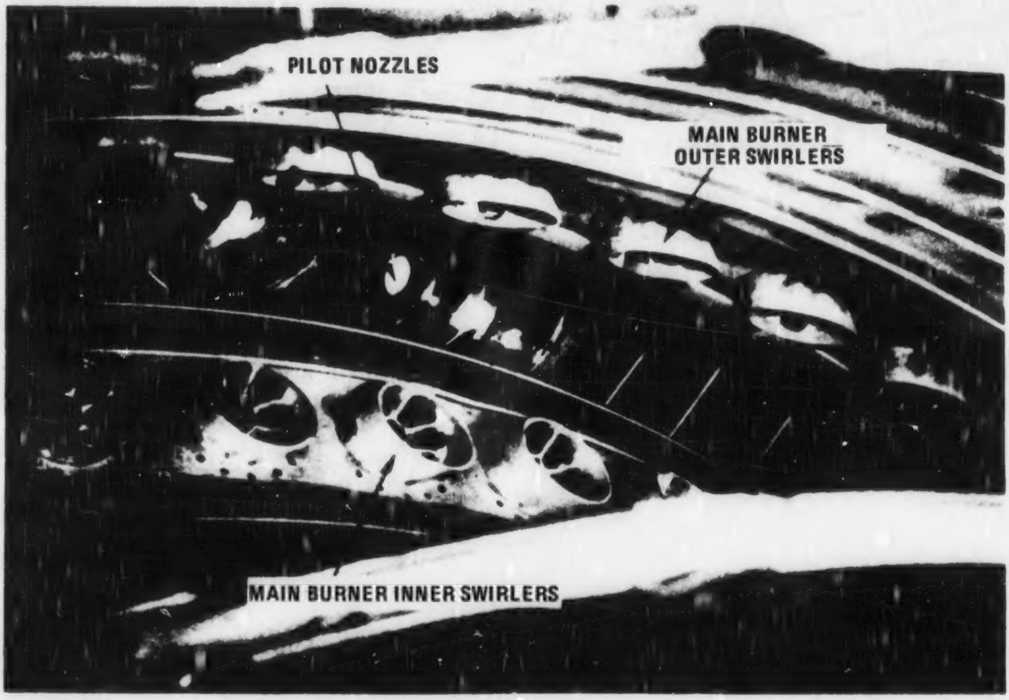

PILOT NOZZLES
MAIN BURNER OUTER SWIRLERS
MAIN BURNER INNER SWIRLERS

Figure 4.49 B. View looking upstream toward headplate showing orientation of main burner inner and outer swirlers relative to the pilot fuel injector nozzle.

Figure 4.49[128]. The Swirl Vorbix Combustor Concept.

TABLE 4.7

SUMMARY FROM TEST RESULTS FOR BEST CONFIGURATIONS OF EACH COMBUSTOR CONCEPT AT SEA-LEVEL TAKE-OFF CONDITION[129]

Combustor Concept and Configuration	Oxides of Nitrogen Emission Index Corrected to Engine Design Take-off Conditions (g/kg fuel)	Percent Reduction from JT9D-7 Production Combustor	Combustion Efficiency (Percent)	SAE Smoke Number
Goals	10.0	----	99.0	15
JT9D-7 Combustor	31.5	----	99.99	10
Swirl Can Configuration N9	13.6	56.8	99.5	1
Staged Premix Configuration P3	20.6	34.6	99.4	6
virl Vorbix Configuration S10	12.4	60.6	99.7	14

NOTES: Oxides of Nitrogen data corrected to engine conditions with combustor inlet pressure of 21.7 atm, combustor inlet temperature of 768.9 K, and combustor inlet airflow rate of 92.9 kg/s.

Combustion efficiency (based on gas sample) and SAE Smoke Number are recorded data at test rig conditions with inlet pressure of 6.8 atm, inlet temperature of 768.9 K, and inlet airflow rate of 6.88 kg/s.

JT9D-7 combustor with the swirl vorbix configuration exhibiting the greatest potential for NO_x reduction. The NO_x data shown in the table were extrapolated by Pratt and Whitney to engine conditions.

4.5.1.2.3 Catalytically supported thermal combustion - Catalytically supported thermal combustion is an emerging and promising concept for the reduction of NO_x, CO and HC emissions from gas turbines. Catalytically supported thermal combustion is believed to involve both catalytic and thermal combustion.[130,131] A schematic of a catalytic combustor is shown in Figure 4.50. In the catalytic combustor, fuel and air are premixed prior to introduction to the catalyst core where the reaction begins and combustion is completed at reaction temperatures below 2800°F; well below the threshold for thermal nitrogen fixation.[133,134,135,136] For comparison, the combustor axial temperature gradients for conventional, lean premix and catalytic combustor concepts are shown schematically in Figure 4-51.

In 1973, Westinghouse, in a joint effort with Engelhard Industries, performed a series of tests to determine the feasibility of using combustion catalysts in stationary gas turbines. Table 4-8 summarizes the results of testing a conventional combustor and a catalytic combustor in the same test rig for combustion of coal gas and Number 2 distillate fuels. The catalytic combustor, as compared to the conventional combustor, reduced NO_x emissions by more than 98 and 89 percent when burning distillate oil and coal gas, respectively, with concurrent reductions in CO and HC emissions. It should be noted that these reductions were accomplished with the catalytic combustor operating at a higher outlet temperature than the conventional combustor. Table 4-9 shows results of additional testing in 1974 and compares emissions from a lean burn premix combustor to those of the catalytic combustor. The catalytic combustor again had very low emissions which were of about the same order of magnitude as the emissions from the premix configuration.

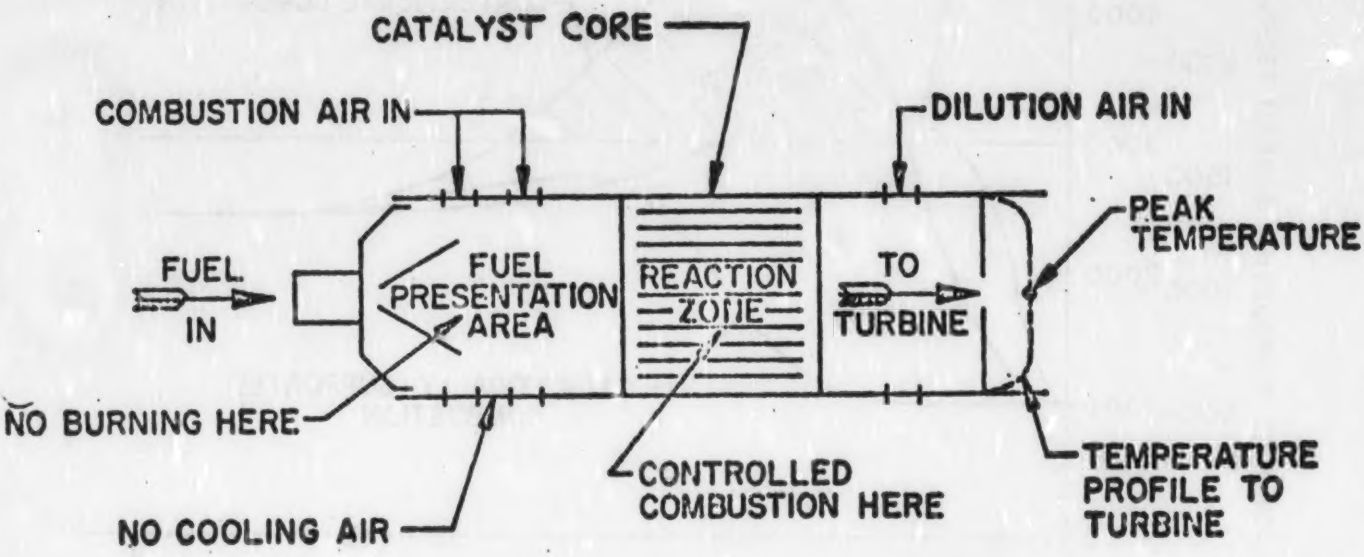

CATALYST CORE
COMBUSTION AIR IN
DILUTION AIR IN
PEAK TEMPERATURE
FUEL IN
FUEL PRESENTATION AREA
REACTION ZONE
TO TURBINE
NO BURNING HERE
CONTROLLED COMBUSTION HERE
TEMPERATURE PROFILE TO TURBINE
NO COOLING AIR

FIGURE 4-50.[132] Catalytic Combustor Design Concept

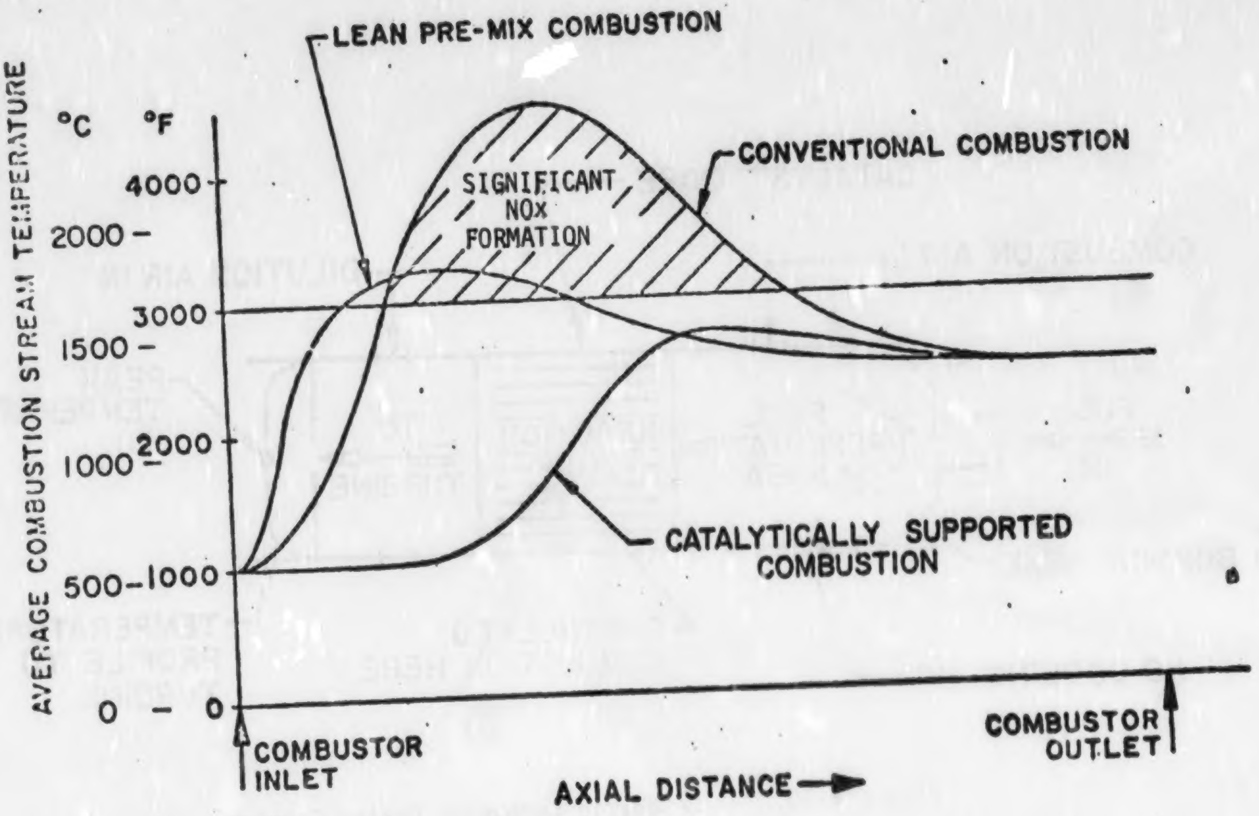


FIGURE 4-51.[137] Temperature History for Conventional Combustors, Lean Pre-Mix Combustors, and Catalytically Supported Combustion

Table 4-8[138] 1973 Test Results from the Westinghouse/Engelhard Catalytic Combustor Evaluation

COMBUSTOR TYPE	FUEL	OUTLET TEMP.		COMBUSTOR PRESS.		NO	CO	HC_x
		°F	(°C)	PSIA	(KPa)	PPM	PPM	PPM
CONVENTIONAL	# 2 DISTILLATE	1800	(982)	42	(289.6)	60	18	8
	COAL GAS	1742	(950)	51.1	(352.3)	9	160	80
CATALYST	# 2 DISTILLATE	2082	(1139)	56.5	(389.6)	<1	10	--
	COAL GAS	2112	(1156)	44.0	(303.4)	<1	40	14

Table 4-9[139] 1974 Test Results from the Westinghouse/Engelhard Catalytic Combustor Evaluation

COMBUSTOR TYPE	FUEL	OUTLET TEMP.		COMBUSTOR PRESS.		NO	CO	HC_x
		°F	(°C)	PSIA	(KPa)	PPM	PPM	PPM
CATALYST	# 2 DISTILLATE	2173	(1189)	47.2	(325.4)	5	30	1
	COAL GAS	2220	(1216)	43.9	(302.7)	1	26	1
PREMIX	# 2 DISTILLATE	1950	(1066)	46.7	(322.0)	6	20	18

Similar low emission levels from catalytic combustion have been verified and reported in tests by Detroit Diesel Allison[140], NASA Lewis Research Center[141], and the Air Force Aero Propulsion Laboratory.[142]

Table 4-10 presents emissions data for catalytically supported thermal combustion when various fuels are burned. Again, the emission levels are extremely low except for the "doped" fuels with high nitrogen content (i.e., propane with 0.17 weight percent nitrogen as ammonia and Number 2 diesel oil with 0.94 weight percent nitrogen as Pyridine). For the "doped" fuels, the conversion of fuel bound nitrogen to NO_x was in the range of 70 to 90 percent.[144] Engelhard presently has research studies underway to use improved catalysts to reduce conversion of fuel nitrogen to NO_x.[145]

Although the results show that catalytically supported thermal combustion is indeed a feasible concept for reduction of emissions from gas turbines, there are still some areas where additional research is required before a production gas turbine using catalytic combustion is a reality. Some of these are:

a. The start-up problem - The catalytic combustor must reach a certain operating temperature, depending on fuel type, for ignition and continued combustion.

b. Variable geometry will be required to operate the catalytic system over the full operating range of a gas turbine.

c. The life of the catalyst and performance over this time period must be determined. Engelhard reports that catalysts in rig tests have demonstrated 1000 hours of operation without degradation in performance.[146] Many stationary gas turbines operate 8000 hours per year.

d. Maintainability and reliability must be demonstrated in a full turbine.

Table 4-10[143] Data for Commercial and Synthetic Fuels with the Engelhard CATCOM DXA-111 Catalyst

	Conditions for Emissions Measurement(4)			Emissions(ppmv)			
Fuel	Inlet Temperature(^{o}C)	Space(5) Velocity (hr.$^{-1}$)	Fuel/Air Ratio (gm/gm)	UHC (as C_3)	CO	NO_x	Remarks
Methane	550	120,000	0.024	10	48	1	(2)
Propane	365	100,000	0.022	3	10	1	(1)
#2 Diesel Oil	380	130,000	0.027	0	48	4	(2)
JP-4	405	180,000	0.022	0	15	1	
Jet A	400	140,000	0.027	6	32	1	(1)
Synthetic Coal Gas (3)	220	300,000	0.313	2	25	1	(1)
Commercial Propane with 0.17 wt. % Nitrogen as Ammonia	250	190,000	0.032	5	10	80	(1)
#2 Diesel Oil with 0.94 wt. % Nitrogen as Pyridine	360	130,000	0.025	3	8	450	(1)

Footnotes:

(1) Stack emissions

(2) Emissions from water cooled sampling probe 2 inches downstream of the catalyst

(3) Fuel composition - (Vol. %): CH_4-3; CO-19, CO_2-9; H_2-15; N_2-54

(4) Total pressure = 1 atm.

(5) Space velocity is a term used to describe combustor loading and is commonly expressed as m^3 gas/m^3 catalyst, hr. = hr^{-1}

4.5.1.2.4 Summary of emission reduction potential of dry control techniques - Table 4.11 is a brief summary of the reductions in NO_x emissions demonstrated in the preceding sections. Depending on the combustor design concept and the type and number of control techniques applied, the percent reduction in NO_x emissions ranged from 12 to greater than 98. Again, one must remember that substantial effort may still be required to incorporate the techniques demonstrated in rig tests into full turbine engines.

Figure 4.52 shows NO_x emissions for tests of gas turbines and combustor rigs using varying degrees and types of dry controls.

4.5.1.3 The Cumulative Effect of Dry and Wet Control Techniques - The preceding sections have quantified reductions in NO_x emissions which have been demonstrated (either on engine tests or rig tests) by the use of wet and dry control techniques. It has been reported, and available data support the conclusion, that dry and wet NO_x controls appear to function independently of each other and their effects are cumulative.[148] This has also been demonstrated by several manufacturers who have incorporated some degree of dry controls into their engines but have still needed to use water injection to meet the more stringent air pollution codes. Figure 4.53 presents NO_x emission data for turbines using wet plus dry control techniques.

4.5.1.4 Catalytic Exhaust Gas Cleanup - Environics has performed research and development work on the catalytic reduction of NO_x by NH_3 in a simulated gas turbine exhaust. Their major conclusion was that NO_x reductions of 80 to 90 percent could be obtained.[149,150] They report that 2000 hours of operating time was recorded for one catalyst, with only minor deterioration in performance.[151]

To our knowledge, this technique has not been demonstrated on a full turbine installation. The laboratory results, however, look very promising. A unique

Table 4-11. Percent Reduction of NO_x Emissions Using Dry Control Techniques

COMPANY	TECHNIQUE	PERCENT REDUCTION FROM PRODUCTION COMBUSTOR
Westinghouse	Lean primary zone and reduced residence time in full size combustor rig	40
	Lean primary in 1/2 size combustor rig	15 - 20
	Exhaust gas recirculation, 1/2 size combustor rig	30 - 38
Solar	Vortex air blast rig test	94(1)
General Electric - NASA Contract	Full size combustor rig tests - lean burn, pre-mix, staged fuel and air	51 and 60(2)
General Electric [147] (Schenectady)	Lean burn, fuel-air mixing	12 - 44
Pratt & Whitney Aircraft NASA Contract	Rig tests; pre-mix, pre-vaporization, staged combustion, lean burn	34.6 - 56.8 - 60.6(2)
Westinghouse and Engelhard	Catalytically supported thermal combustion - rig tests	98

(1) This is a 94 percent reduction from the baseline VAB combustor.

(2) Different combustor concepts

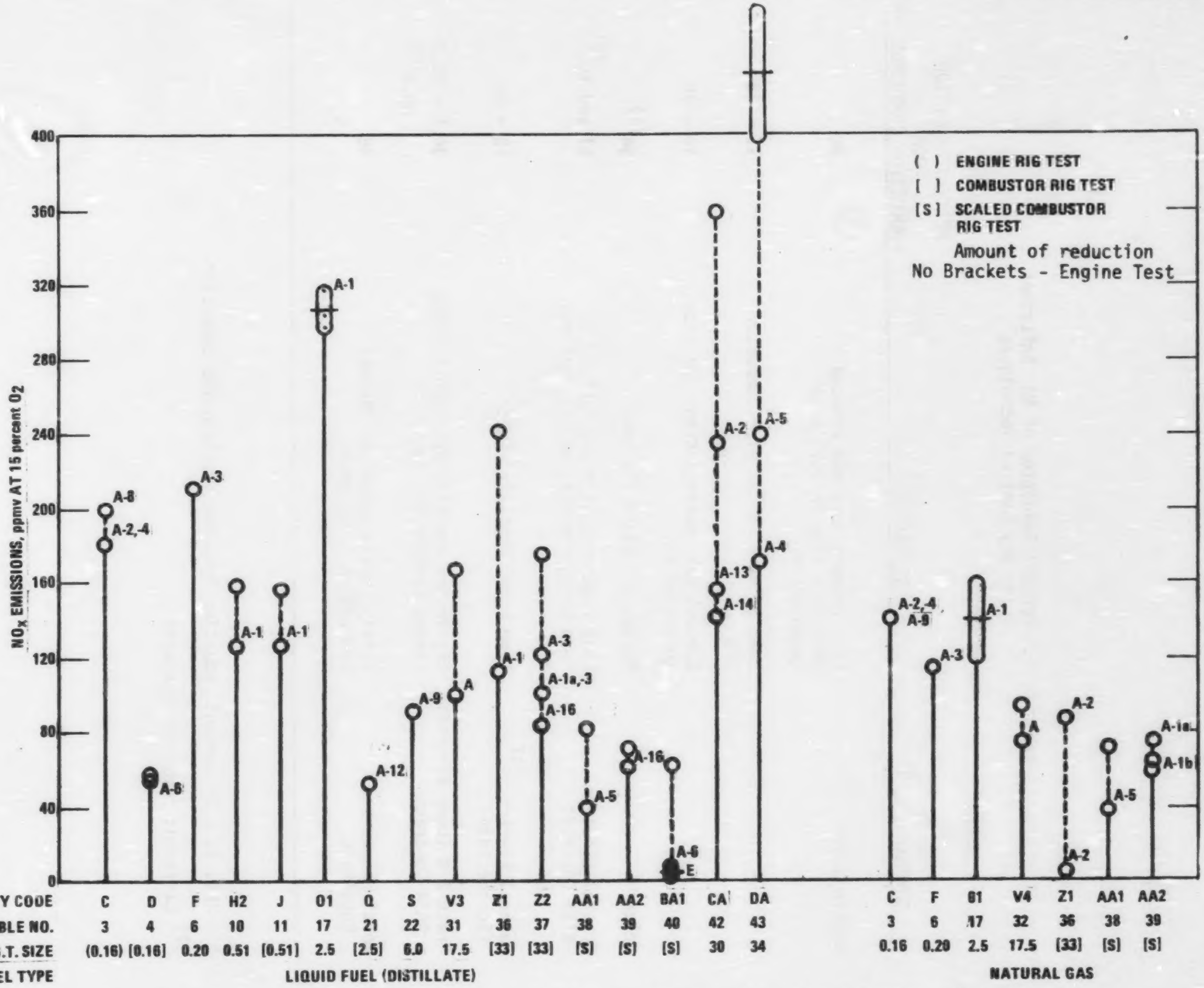


Figure 4.52. Summary of NO_x emission data from gas turbines using dry control techniques.

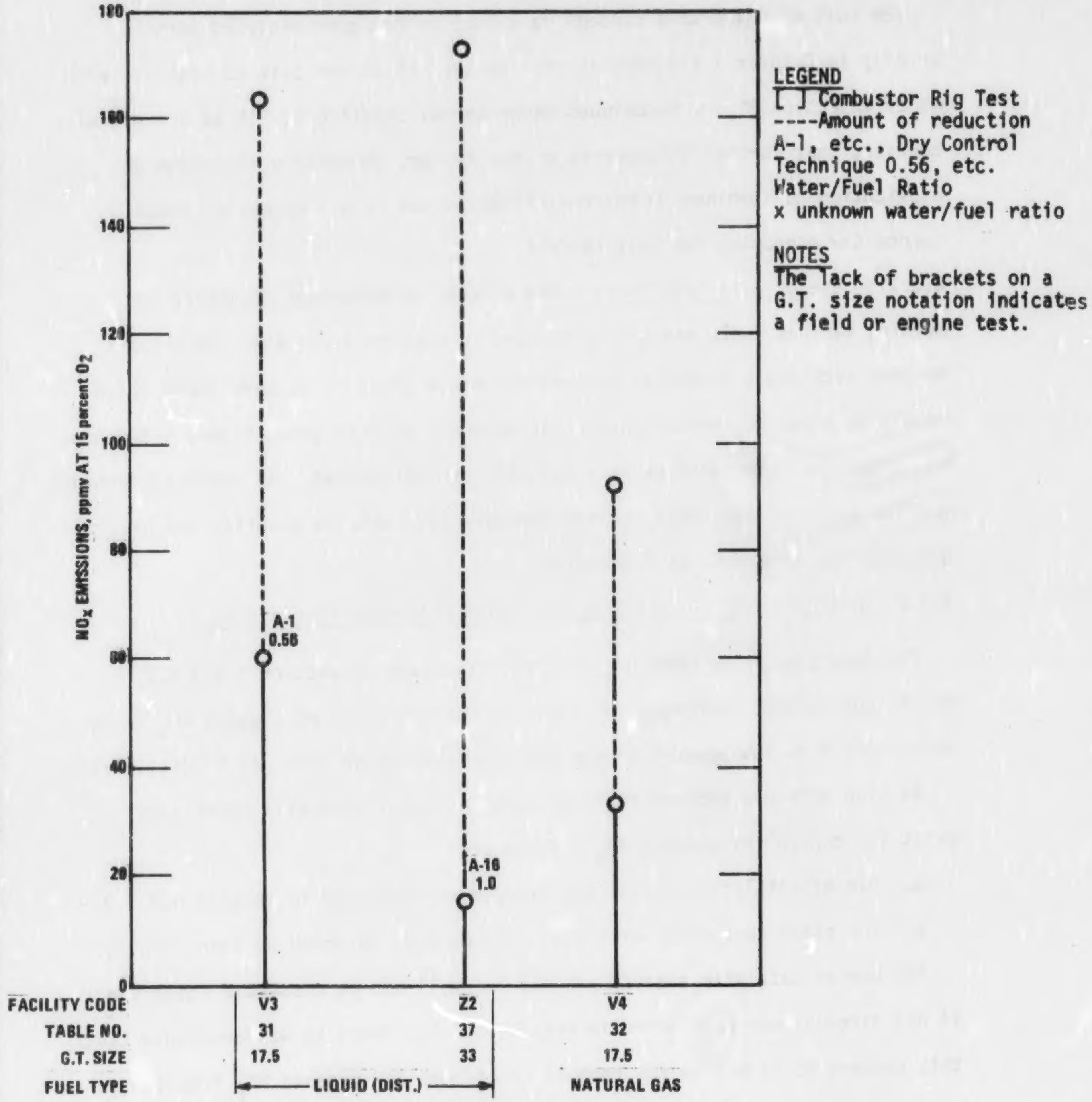


Figure 4.53. Summary of NO$_X$ emissions data from gas turbines using both dry and wet controls.

advantage of this exhaust gas catalytic cleanup device is that it could be used for a gas turbine burning fuels with high bound nitrogen contents to remove the NO_x formed from fuel bound nitrogen and thermally formed NO_x.

The cost of using this concept on a turbine has been analyzed very briefly in Chapter 7 and indications are that it is not cost competitive with water injection or dry techniques under normal conditions. It is conceivable, however, that further refinements of the system, production in volume or application to turbines in extreme circumstances (e.g., no water) could change the economics for this concept.

4.5.1.5 Use of Alternate Fuels - The effects on emissions of thermal NO_x burning various fuels has been discussed in section 3.2.3.4.3. As shown in that section, a potential control technique would be to burn fuels which result in lower NO_x emissions such as methanol or coal gas. Figure 4-54 shows NO_x emissions from turbines as a function of fuel burned. At present, methanol and low Btu coal gas fuels are not commonly available in quantity and natural gas supplies appear to be decreasing.

4.5.2 Control of NO_x Formed from Fuel Bound Nitrogen (Organic NO_x)

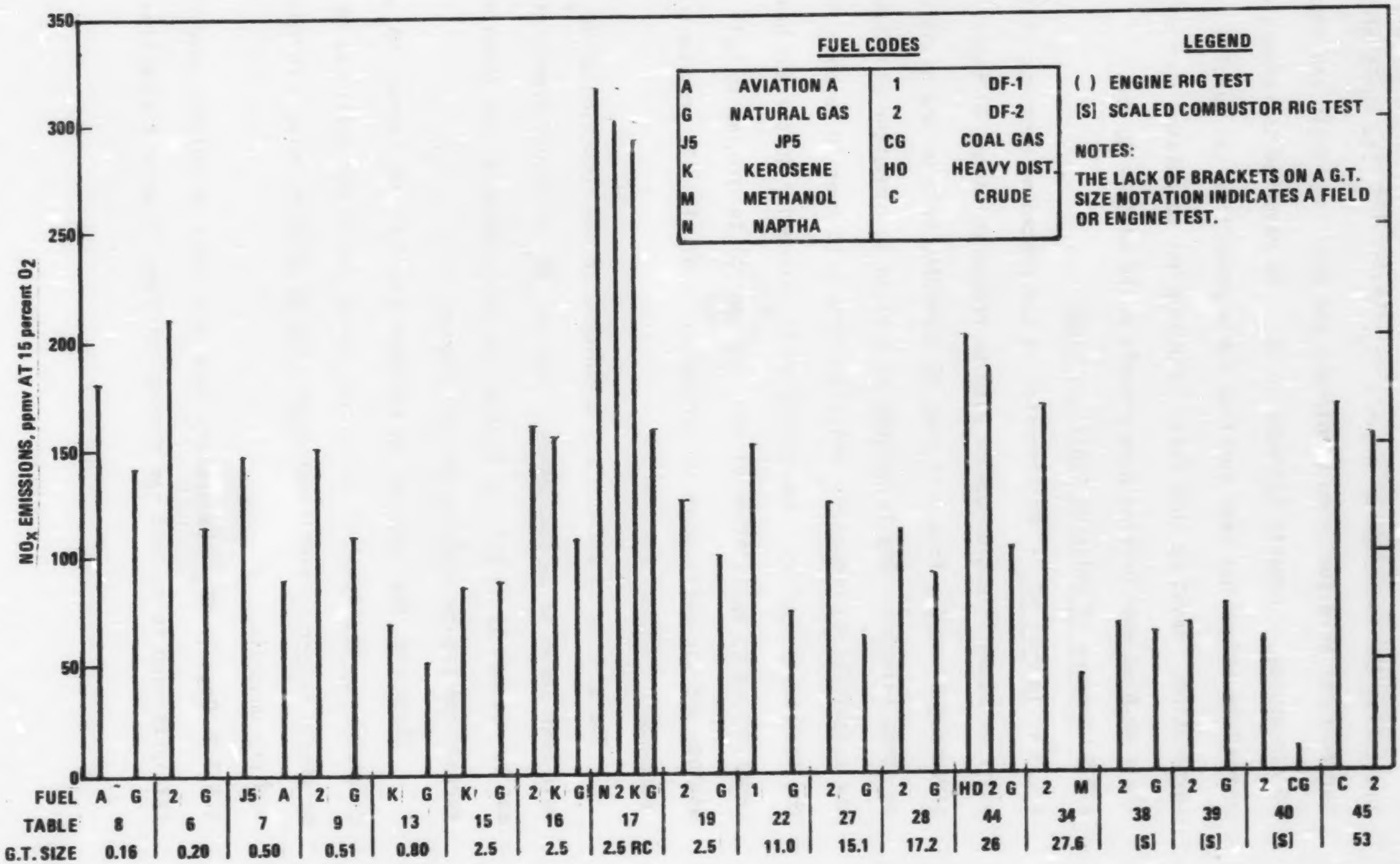
The formation of organic NO_x has been discussed in section 3.2.3.4.2. An obvious control technique for limiting the emissions of organic NO_x is to burn fuels with low bound nitrogen contents such as natural gas or distillates.

If high nitrogen content fuels are used, several potential techniques exist for control of organic NO_x. These are:

a. Use of catalytic exhaust gas cleanup as discussed in section 4.5.1.3.

b. Two stage combustion with fuel rich burning followed by lean burning.

The use of catalytic exhaust gas cleanup will not be discussed again since it has already been considered in section 4.5.1.3 where it was concluded that this concept is effective for removal of thermal and organic NO_x from the exhaust stream.


NO$_X$ EMISSIONS, ppmv AT 15 percent O$_2$
350
300
250
200
150
100
50
0
FUEL
TABLE
G.T. SIZE
A G
8
0.16
2 G
6
0.20
J5 A
7
0.50
2 G
9
0.51
K G
13
0.80
K G
15
2.5
2 K G
16
2.5
N 2 K G
17
2.5 RC
2 G
19
2.5
1 G
22
11.0
2 G
27
15.1
2 G
28
17.2
HD 2 G
44
26
2 M
34
27.6
2 G
38
[S]
2 G
39
[S]
2 CG
40
[S]
C 2
45
53
FUEL CODES
A AVIATION A
G NATURAL GAS
J5 JP5
K KEROSENE
M METHANOL
N NAPTHA
1 DF-1
2 DF-2
CG COAL GAS
HO HEAVY DIST.
C CRUDE
LEGEND
( ) ENGINE RIG TEST
[S] SCALED COMBUSTOR RIG TEST
NOTES:
THE LACK OF BRACKETS ON A G.T. SIZE NOTATION INDICATES A FIELD OR ENGINE TEST.

Two stage combustion has been reported as an effective method of preventing the formation of organic NO_x.[152,153,154] The fuels are essentially burned first in a fuel rich atmosphere and then in a fuel lean atmosphere with no conversion of organic nitrogen to NO_x. As reported in reference 154, this control technique has been verified for a production gas turbine by Turbodyne Corporation. Based on this data, Turbodyne has calculated that emissions from their machine when burning pure ammonia would be only 30 ppm.

4.5.3 Summary of Emission Control Systems

4.5.3.1 Thermal NO_x - As discussed in the preceding sections of this chapter, water or steam injection can be used to reduce emissions of themal NO_x by percentages varying from 0 to over 80 depending only on the quantities of water or steam injected. Dry techniques as applied to combustor rigs have demonstrated the capability to reduce NO_x emissions from 0 to over 90 percent, depending on the techniques applied. Reductions in NO_x exceeding 40 percent have been demonstrated by application of some dry control techniques to full turbine engines and the application of further dry controls will provide additional reductions.

The effects of wet and dry techniques on the reduction of NO_x emissions have been shown to be cumulative. That is, NO_x emissions from a given engine may be reduced by 40 percent through the application of dry techniques and then further reduced by applying wet techniques.

Therefore, the level of an emission standard for thermal NO_x will not be determined by the limit of technology since almost any level can be obtained by simply injecting additional quantities of water or using combinations of wet and dry controls.

4.5.3.2 Organic NO_x - Laboratory tests and tests on boilers have shown that catalysts used in exhaust gas streams are over 70 percent effective in removing

NO_x, regardless of source. Also, as discussed earlier in this chapter, two stage combustion (rich combustion followed by lean combustion) can be used to eliminate formation of organic NO_x. This has been demonstrated in a production turbine engine marketed by Turbodyne Corporation. A third technique is to burn low nitrogen fuels.

In any case, gas turbines do not commonly burn the high nitrogen fuels. Also, as discussed in section 4.3, there does not appear to be any present trend towards the combustion of those fuels in gas turbines. Since the contribution of organic NO_x from a turbine burning a high nitrogen fuel can exceed the thermal NO_x, a gas turbine regulation to control NO_x emissions should regulate both forms of NO_x.

4.5.3.3 HC and CO - As discussed in Chapters 3 and 4, HC and CO emissions from gas turbines are not significant for full load operations. However, emissions of both pollutants increase with decreasing load. Wet control techniques do not reduce CO or HC emissions. Several dry control techniques, however, have been demonstrated for reducing the CO and HC emissions over the turbine operating range.

Most gas turbines are operated only at or near full load conditions. Some utilities, however, operate gas turbines in a low power "spinning reserve" mode so that they can be rapidly brought up to full power in case of frequency, voltage or power fluctuations. For example, a 67 MW turbine operating in spinning reserve emits about 960 pounds per hour of CO. Since turbines are commonly used in clusters, it is obvious that a potential exists for significant impact on local ambient air quality. This impact is considered in Chapter 6.

4.5.3.4 SO_2 - As discussed in section 4.3, the emissions of SO_2 from gas turbines are strictly a function of the sulfur content of the fuel burned and flue gas scrubbing for SO_2 removal is impractical (compared to the cost of desulfurizing the fuel) because of the large volumes of gas which must be treated. The relative costs are briefly discussed in Chapter 7. Therefore, a standard for SO_2 emissions would essentially limit the sulfur content of the fuel which can be burned and would depend on factors other than control techniques applicable to the turbine.

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5. MODIFICATION AND RECONSTRUCTION

In accordance with Section 111 of the Clean Air Act, as amended in 1970 and 1974, standards of performance shall be established for new sources within a stationary source category which ". . . may contribute significantly to air pollution . . ." Standards apply to operations or apparatus (facilities) within a stationary source, selected as "affected facilities", that is, facilities for which applicable standards of performance have been promulgated and the construction or modification of which commenced after the proposal of said standards.

On December 16, 1975, the Agency promulgated amendments to the general provisions of 40 CFR Part 60, including additions and revisions to clarify modification and the addition of a reconstruction provision. Under the provisions of 40 CFR 60.14 and 60.15, an "existing facility" may become subject to standards of performance if deemed modified or reconstructed. An "existing facility" defined in 40 CFR 60.2(aa) is an apparatus of the type for which a standard of performance is promulgated and the construction or modification of which was commenced before the date of proposal of that standard. The following discussion examines the applicability of these provisions to stationary gas turbine facilities and details conditions under which existing facilities could become subject to standards of performance. It is important to stress that since standards of performance apply to affected facilities, which, combined with existing and other facilities comprise a stationary source, the addition of an affected facility to a stationary source through any mechanism, new construction, modification, or reconstruction, does not make the entire stationary source subject to standards of performance, only the added affected facility.

5.1 40 CFR PART 60 PROVISIONS FOR MODIFICATION AND RECONSTRUCTION

5.1.1 Modification

It is important that these provisions be fully understood prior to investigating their applicability.

Section 60.14 defines modification as follows:

> "Except as provided under paragraphs (d), (e) and (f) of this section, any physical or operational changes to an existing facility which result in an increase in emission rate to the atmosphere of any pollutant to which a standard applies shall be a modification. Upon modification, an existing facility shall become an affected facility for each pollutant to which a standard applies and for which there is an increase in the emission rate".

The exception in paragraph (d) allows for an existing facility to undergo a physical or operational change which results in an increase in the emission rate of any pollutant to which a standard applies, but not be deemed a modification, provided the owner or operator can demonstrate to the Administrator's satisfaction that the total emission rate of that pollutant from all facilities within the stationary source to which an appropriate EPA reference method or alternative method listed in paragraph (b) can be applied has not increased. The required reduction in emission rate may be accomplished through the installation or improvement of a control system or through physical or operational changes to facilities including reducing the production of a facility or closing a facility.

Paragraph (e) lists certain physical or operational changes which will not be considered as modifications, irrespective of any change in the emission rate. These changes include:

1 - Routine maintenance, repair, and replacement.

2 - An increase in the production rate not requiring a capital expenditure as defined in Section 60.2(bb).

3 - An increase in the hours of operation.

4 - Use of an alternative fuel or raw material if prior to the standard, the existing facility was designed to accomodate that alternate fuel or raw material.

5 - The addition or use of any system or device whose primary function is the reduction or air pollutants, except when an emission control system is removed or replaced by a system considered to be less efficient.

Paragraph (b) clarifies what constitutes an increase in emissions in kilograms per hour and the methods for determining the increase, including the use of emission factors, material balances, continuous monitoring systems, and manual emission tests. Paragraph (c) affirms that the addition of an affected facility to a stationary source does not make any other facility within that source subject to standards of performance. Paragraph (f) simply provides for superceding any conflicting provisions.

5.1.2 Reconstruction

Section 60.15 regarding reconstruction states:

> "If an owner or operator of an existing facility proposes to replace components, and the fixed capital cost of the new components exceeds 50 percent of the fixed capital cost that would be required to construct a comparable entirely new facility, he shall notify the Administrator of the proposed replacements. The notice must be postmarked 60 days (or as soon as practicable) before construction of the replacements is commenced. . ."

The purpose of this provision is to ensure that an owner or operator does not perpetuate an existing facility by replacing all but vestigial components, support structures, frames, housings, etc., rather than totally replacing it in order to avoid subjugation to applicable standards of performance.

5.2 APPLICABILITY TO STATIONARY GAS TURBINE INSTALLATIONS

5.2.1 General[1, 2, 3, 4]

In the gas turbine industry, the terms "overhaul" and "ultimate life" are not definable in terms used equally by all manufacturers. As an example, one manufacturer's "overhaul" is another manufacturer's "major disassembly inspection". To compound this nebulous situation, the intervals called "overhaul life" and "ultimate life" vary significantly among manufacturers and users. The variations in "overhaul life" and "ultimate life" are influenced by differences in gas turbine applications, operating loads, operating hours per start, fuel quality and maintenance practices. Most turbines are designed for 20,000 to 40,000 operating hours between overhauls and for an ultimate life of about 30 years. The 30 year figure for ultimate life is the design life of non-wearing parts such as rotor discs, casings, housing, bearing supports, and rotor shafts. Typically an overhaul returns a unit to a "new" condition through the replacement of wearing or working parts and, therefore, the "ultimate life" of a gas turbine may be indefinite.

5.2.2 Modification

Physical and operational changes to a gas turbine which might be considered a modification are:

a. Replacement of components with a different design than the original to permit firing a turbine with fuels for which it was not originally designed, such as crude or residual oils.

b. Replacement of components with a different design than the original to increase the power output of the turbine.

c. Sustained operation of a turbine at higher outputs than design (this can be performed at a sacrifice in turbine life).

It is highly unlikely that an owner or operator would opt to perform items a, b, or c above because:

a. As discussed in Chapter 4, there does not appear to be any trend to use crude or residual oils in gas turbines.

b. Power output of a turbine is commonly increased by water or steam injection rather than replacement of components

c. Gas turbine life decreases rapidly when operation occurs above design levels for sustained periods.

As indicated in Chapter 3, stationary gas turbines have many uses and may be located at oil refineries, power plants, pumping stations, etc. When a gas turbine is located in a plant which has other sources of the same pollutants generated by turbines (e.g., refineries, power plant complexes, etc.), then increased pollutants due to a modification of the turbine can be offset by the compensatory emission reduction allowed under paragraph (d) of Section 60.14. This compensatory emission reduction must be accomplished by upgrading an existing control system, by adding a new control system, or by making physical or operational changes on facilities within the source which are amenable to conventional emission tests methods so that the owner or operator can demonstrate by emission tests that no net increase in emissions has occurred. The exception to the emission testing requirement would occur if the compensatory emission reduction resulted from an operational change such as reducing the production capacity or shutting down a piece of process equipment to which an acceptable emission factor applied.

The following physical or operational changes will not be considered as modifications to existing gas turbines:

a. Changes determined to be routine maintenance, repair, or replacement in kind. This will include repair or replacement of stator blades, turbine nozzles, turbine buckets, fuel nozzles, combustion chambers, seals, and shaft packings.

b. Changes in type or grade of fuel used, if the original gas turbine installation, fuel nozzles, etc., were designed for its use.

c. An increase in hours of operation.

d. Variations in operating loads within the engine design specification. As noted in Chapter 3, emissions from gas turbines vary with operating loads.

e. The relocation or change in ownership of an existing facility.

The impact of the modification provision on existing gas turbines should be very slight.

5.2.3 Reconstruction

A reconstructed turbine, as discussed in 5.1.2, is essentially a turbine which has undergone a major rebuilding when it would otherwise have been scrapped or recycled. It is difficult to apply the definition of reconstruction to a gas turbine because substantial portions of a turbine may be replaced as a matter of routine maintenance during the normal overhauls as described in 5.2.1. Since it is current practice to replace substantial portions of turbines, it would be difficult to discriminate between a major overhaul that was performed to avoid the purchase of a new turbine and one that was performed in accordance with a routine maintenance program. Such routine maintenance should be exempted from the regulatory consequences of becoming a reconstructed turbine, subject to the "50 percent rule" discussed in 5.1.2.

REFERENCES FOR CHAPTER 5

1. Letter and attachments, Sievert, M. O., Solar, to Don R. Goodwin, EPA, ESED. p. 4. November 10, 1975.
2. Letter and attachments, Gaylord, R. H., Turbodyne, to Don R. Goodwin, EPA, ESED, Section C, December 19, 1975.
3. Letter and attachments, Zeltmann, E., General Electric, to Kenneth R. Durkee, EPA, ESED. p. 22.
4. "Gas Turbine Maintenance". R. H. Knorr. General Electric. pp. 1-13. 1974.

6. ENVIRONMENTAL IMPACT

As discussed in section 3.2.3.5, only a few state and local jurisdictions have regulations specifically for stationary gas turbines, although some have applied standards originally written for other sources to gas turbines. Therefore, for purposes of this chapter, the environmental impact of the emission control techniques will be discussed by comparing their impact with the impact assuming gas turbines are uncontrolled.

6.1 AIR POLLUTION IMPACT

6.1.1 Introduction

This section evaluates the air quality impact of different levels of control for SO_2, CO, and NO_x emissions from gas turbines. Through the use of dispersion models, concentration estimates are made for each of the three pollutants for averaging times consistent with National Ambient Air Quality Standards (NAAQS).[1] These standards and averaging times are tabulated below for reference.

		Standard $\mu g/m^3$	
Pollutant	Averaging Time	Primary	Secondary
NO_2	Annual	100	100
CO	8 hr.	10 mg/m^3	10 mg/m^3
	1 hr.	40 mg/m^3	40 mg/m^3
SO_2	Annual	80	
	24 hr.	365	
	3 hr.		1300

Dispersion analyses for nine stationary gas turbines operating singly and in clusters and for operation at full load and "spinning reserve" are considered. For these analyses, the effects of varying the sulfur content of the fuel from 0.01 to 2.7 percent are investigated. Uncontrolled levels of CO emissions and alternative controlled levels of 90 ppm, 125 ppm, 250 ppm and the estimated uncontrolled level at spinning reserve of 500 ppm are analyzed. The effect on ambient air quality of uncontrolled levels of NO_x emissions and alternative controlled levels of 75 ppm, 100 ppm, and 125 ppm is also analyzed. As discussed in Chapter 3 and 4: (a) SO_2 emissions are controlled by limiting the sulfur content of the fuel used; (b) CO emissions are a function of combustor geometry and turbine efficiency; (c) NO_x can be controlled to almost any level by water injection and dry control techniques.

The dispersion analyses also considers the effects on ambient air quality levels of varying wind speeds and variations in exhaust stack height.

For the purposes of these analyses, all three pollutants are assumed to behave in the atmosphere as non-reactive gases. All NO_x emissions and concentrations are assumed to be equivalent to those for NO_2. The estimated pollutant concentrations are based on the application of state-of-the-art modeling techniques, which implies reliability of the estimates to within a factor or two.

6.1.2 Source Characteristics

The major emphasis in these analyses is an assessment of the maximum potential air quality impact of SO_2, CO, and NO_x emissions for both individual and cluster arrangements of gas turbines. Source data for the nine individual stationary gas turbine units are reported in Table 6-1. The difference in emission levels for the gas turbine unit is proportional to their levels of power output. Thus, it can be expected that the individual air quality impact for the larger gas turbine units may be greatest.

Table 6 - 1
Stack Parameters

Manufacturer & Model	Power Output (MW)	Cycle Type	H_s (m)	T_s (deg.K)	V_s (m/sec)	D_s (m)	H_b (m)
Garret Airesearch							
1. GTC8S-90 (GPUT-90)	0.158	Simple	1.9	903	57	0.3	5
Solar							
2. Saturn	0.8	Simple	4.6	733	77	0.45	3.69
3. Centaur	2.5	Simple	4.9	700	68	0.75	3.38
General Electric							
4. MS5001P (Spinning Reserve Mode)	24.2	Simple	10.8	754 (510)	19.2 (12.2)	4.12	7.9
5. MS5002B	23.5	Simple	6.4	622	15.7	3.89	7.9
6. MS7001B (Spinning Reserve Mode)	60.3	Simple	9.7	780 (528)	33.7 (20.7)	4.44	8.69
7. MS7001B (Two Machines)	161.6	Unfired Combined	15.9	444	27.4	5.33 (Double Unit)	8.69
		By-Pass Stack	12.8	780	0.38	4.31 (Each Unit)	
8. MS7001B (Spinning Reserve Mode)	57.4	Regenerative	9.4	639 (432)	23.7 (15.1)	3.44	8.69
9. MS7001C (Spinning Reserve Mode)	67.4	Simple	9.7	811 (549)	38.5 (24.5)	4.46	8.69

H_s = stack height
T_s = stack temperature
V_s = stack exit speed
D_s = equivalent stack diameter
H_b = height of highest nearby structure

Turbine units number 4 through 9 have rectangular shaped stacks so the diameter is based on an equivalent area circle. Stack velocity is based on the mass flow rate and stack exit area. Turbine units number 4, 6, 8, and 9 additionally are considered in a spinning reserve mode where the exit temperature and velocity are reduced.

The cluster arrangements for which analyses are made are shown in Figure 6-1. These arrangements represent expected groupings of the gas turbines.

6.1.3 Dispersion Models

The dilution of effluents from stationary gas turbine units is much different from that of sources with conventional tall round stacks. For many gas turbines the effluent stacks are not much higher than the nearby building structure which houses the turbines. The short effluent stacks may adversely affect both plume rise and dispersion. In addition, many of the effluent stacks are unconventionally rectangular in shape and some are stuffed with acoustic baffling which results in a nonuniform effluent exit stream. Under high wind conditions these stack characteristics can be expected to frequently result in a severe effect on dispersion of the effluent due to entrainment of the plume into the stack wake. The adverse affects of the turbine structure and stacks on plume rise and effluent dispersion generally result in higher concentrations than those associated with a source having similar emission rates, but more conventional characteristics.

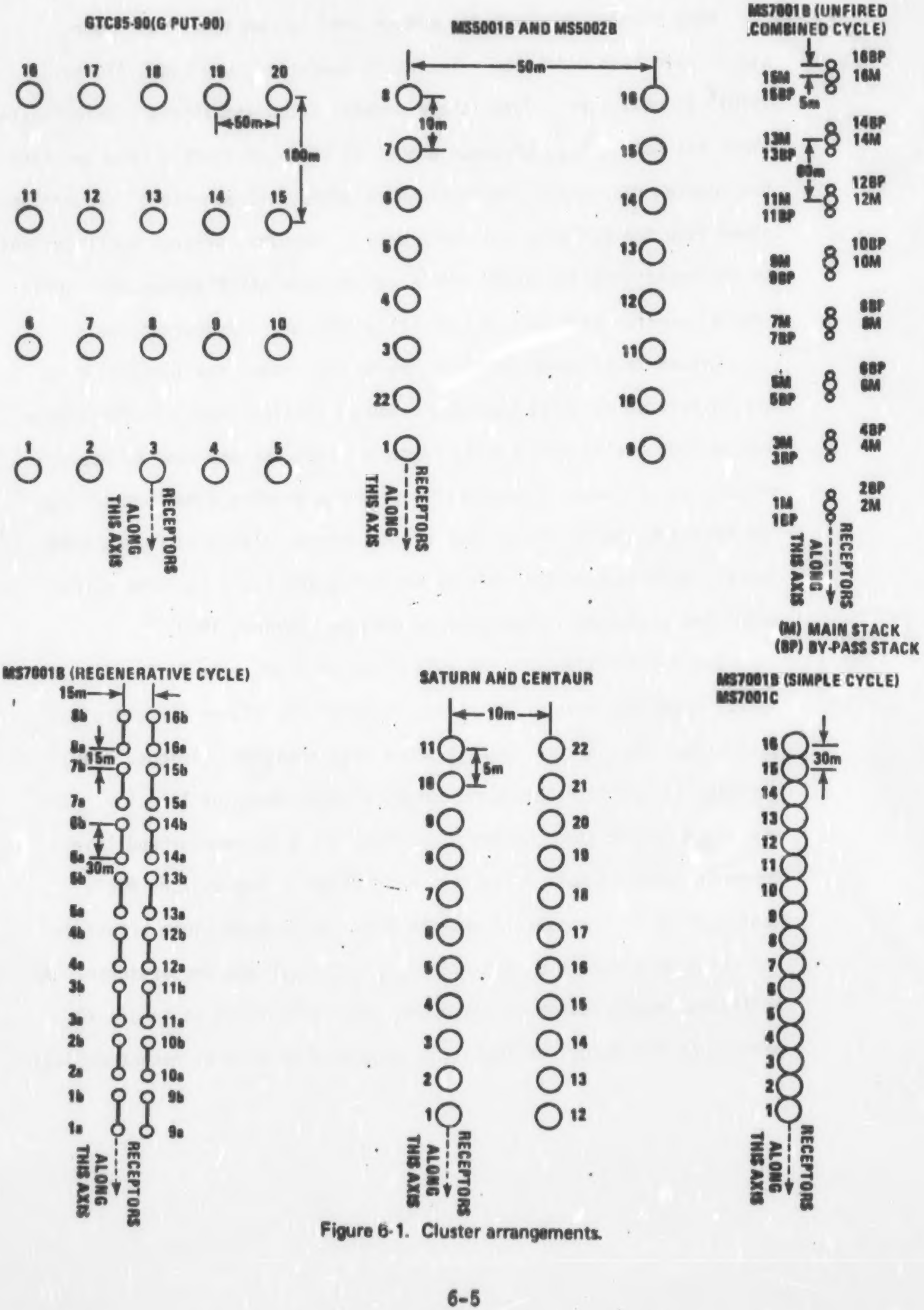
GTC85-90(G PUT-90)
16
17
18
19
20
50m
100m
11
12
13
14
15
6
7
8
9
10
1
2
3
4
5
RECEPTORS ALONG THIS AXIS
MS5001B AND MS5002B
50m
10m
8
7
6
5
4
3
22
1
16
15
14
13
12
11
10
9
RECEPTORS ALONG THIS AXIS
MS7001B (UNFIRED COMBINED CYCLE)
16BP
16M
15M
15BP
5m
14BP
14M
13M
13BP
60m
12BP
12M
11M
11BP
10BP
10M
9M
9BP
8BP
8M
7M
7BP
6BP
6M
5M
5BP
4BP
4M
3M
3BP
2BP
2M
1M
1BP
RECEPTORS ALONG THIS AXIS
(M) MAIN STACK
(BP) BY-PASS STACK
MS7001B (REGENERATIVE CYCLE)
15m
8b
8a
15m
7b
7a
6b
6a
30m
5b
5a
4b
4a
3b
3a
2b
2a
1b
1a
16b
16a
15b
15a
14b
14a
13b
13a
12b
12a
11b
11a
10b
10a
9b
9a
RECEPTORS ALONG THIS AXIS
SATURN AND CENTAUR
10m
5m
11
10
9
8
7
6
5
4
3
2
1
22
21
20
19
18
17
16
15
14
13
12
RECEPTORS ALONG THIS AXIS
MS7001B (SIMPLE CYCLE)
MS7001C
30m
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
RECEPTORS ALONG THIS AXIS

Figure 6-1. Cluster arrangements.

Many studies of plume dispersion from conventional tall round stacks have been conducted. The basic Gaussian plume model (Turner, 1970)[2] is well accepted for those uncomplicated situations. Unfortunately, there has not been an adequate amount of study to thoroughly understand the complex dispersion processes which affect entrainment of the effluent plume into the building and stack wakes. However, several modifications to the basic Gaussian plume models can be made which approximate these general adverse effects. The modifications are discussed below.

Ground level concentrations due to individual gas turbine units are calculated by using the Single Source (CRSTER) Model.[3] The Single Source Model calculates hourly concentrations for one year of hourly meteorological data. Concentrations over averaging times longer than one hour are simply the average of the values calculated during those hours. Calculations are made by employing the basic Gaussian plume model and associated dispersion parameters (Turner, 1970).[2]

Two modifications to the estimation of plume rise in the Single Source Model are made which result in generally higher concentration estimates. One, Briggs bouyant plume rise equations (Briggs, 1969)[4] are used to estimate the plume height at each receptor location. The use of the final plume height is avoided since maximum ground level concentrations in most instances occur close to the source. This modification is necessary since the final calculated plume height is very high as a result of the extremely hot stack gas temperatures. An additional modification to the plume rise calculation is to use 70 percent of the Briggs estimate, as suggested by several recent studies of stationary gas turbines (Hoult and O'Dea,[5] 1975, Egan, et. al.,[6] 1975). This accounts for reduced plume rise resulting from strong wind shear and increased turbulence in the plume due to nearby buildings.

Dispersion of effluents emitted from short stacks is affected by increased turbulence induced by nearby buildings. A recent study (Huber and Snyder, 1976)[7] has suggested that the adverse effects that buildings can have on plume entrainment are best accounted for by enhancing the vertical dispersion parameter. Thus, the following evaluation is made to determine if the building wake adversely affects the effluent plume. The plume centerline height at a distance of 2 building heights downwind from the turbine structure is estimated. For those instances where the estimated height is less than 2.5 times the height of the nearby turbine structure, the conventional vertical dispersion parameters are enhanced following the suggestions of Huber and Snyder (1976).[7] This modification results in higher estimated concentrations close to the source.

Ground level concentrations are calculated with the modified Single Source Model at receptors placed at 10 degree intervals at downward distances of 0.1, 0.2, 0.5, 2.0 and 20.0 kilometers. Meteorological data for Dallas, Texas for the year 1964 are used in the analyses. Dallas, Texas is selected as a site which is not complicated by terrain and has mean wind speeds rather typical of many moderately windy locations throughout the United States.

In addition to the dispersion analyses for individual gas turbines, analyses of turbine unit clusters are conducted. The Single

Source Model does not consider multiple source locations. Thus, the PTMTP (UNAMAP) Model[8] is used for the cluster analyses. The model calculates hourly concentrations for up to twenty-five point sources. Again, the basic Gaussian plume model is used with Briggs plume rise estimations. Concentrations for a number of hours up to 24 can be calculated, and an average concentration over any number of hours can be additionally obtained. Modifications to plume rise and an enhancement of the vertical dispersion parameters similar to those made to the Single Source Model are made to PTMTP for these analyses.

For the cluster arrangements, concentrations are estimated at distances of 0.1, 0.2, 0.5, 2.0 and 20 kilometers from the downwind edge of the cluster.

Cluster analyses are conducted with PTMTP by using the meteorological conditions associated with the period for which the highest ground level concentrations are estimated by the Single Source Model. Also, the wind directions are aligned along the major axis of the cluster arrangement. This technique provides an indication of the maximum potential impact. However, it is somewhat limited in that it does not provide detail as to the variability of concentrations throughout the year, as was possible with the Single Source Model.

Plume rise from individual units in the cluster of turbines is considered in the same manner as for the single source analysis. However, there may be some reason to believe that interaction of the plumes from the multiple sources results in enhanced plume rise

(Milhous et. al., 1975).[9] Since it is necessary here to determine the maximum potential impact, no enhancement is considered since maximum ground level concentrations occur on windy days where close to the source building wake effects predominate over plume rise. In addition, the highest ground level concentrations are expected to occur close to the source before interaction of many of the sources may be significant.

6.1.4 Analysis of Air Quality Impact

A summary of maximum concentrations found in the individual analysis of all nine gas turbines is presented below. All short-term maximum estimated concentrations occur at 0.1 km except for the Solar (Centaur) turbine which has 24-hour maximums at 0.2 km, and for the G.E. MS7001B turbine which has 1-hour and 3-hour maximums at 0.5 km. The output of the Single Source Model reports the first and second maximum values which occur at each of the thirty six receptors around each ring. The number of occurrences of values which are 90% and 50% of the maximum ground level concentration found among the seventy-two values at the ring distance in which the maximum is found, is also considered.

The highest annual concentrations occurred at 0.1 km except for the Solar (Saturn) turbine which has maximums at 0.5 km, and for the Solar (Centaur) turbine which has maximums at 0.2 km. Since the usage rate for most of the turbines is less than a full year, concentrations lower than those estimated here, likely occur. Annual averages resulting from turbines operating in the spinning reserve mode are considered insignificant. Several levels of the emission rates are used in the calculations for cluster arrangements. For those situations having all turbines operating in the same mode, concentration estimates for other emission rates change proportionately. Note that under the spinning reserve mode of operation SO_2 emissions are assumed to be one-third those under full load.

It should be noted that in all cases NO_x and NO_2 are treated interchangeably for assessing the air quality impact relative to NAAQS.

The estimated air quality impact of the Garrett Airesearch and the Solar gas turbines presented in Tables 6-2 to 6-7 are generally small due to the small pollutant emissions of these sources. For almost all cases considered, even with 22 units operating in a cluster, it is estimated that no NAAQS are exceeded. The one exception is the Centaur (Solar) model turbine. With 22 units operating in a cluster and using greater than 0.3% sulfur (S) fuel, SO_2 NAAQS can be exceeded.

Due to their size the General Electric turbines are characterized by significantly greater emissions and air quality impact than the Garrett Airesearch or the Solar turbines. Five different units are considered below.

The air quality impact of a single MS5001P unit operating in either the regular mode or the spinning reserve mode is not estimated to exceed any NAAQS (Table 6-8). However, it is estimated that the SO_2 3-hour and 24-hour standards can be exceeded by a cluster with

Table 6 - 2
Summary of Maximum Ground Level Concentrations for Garret Airesearch Gas Turbine, Model GTC8S-90 (GPUT-90)

SO_2 Emissions S Fuel)	(g/sec)	3-Hour SO_2 Average ($\mu g/m^3$)	24-Hour SO_2 Average ($\mu g/m^3$)	Annual SO_2 Average ($\mu g/m^3$)
Jet A Fuel				
0.01	0.005	1.8	1.2	0.1
0.1	0.05	17.8	12.4	1.2
0.3	0.14	49.7	34.8	3.6
0.8	0.38	134.0	94.4	9.6
Number of Periods within stated percent of maximum				
90%		9	1	
50%		55	9	
CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m^3)	8-Hour CO Average (mg/m^3)	
Jet A Fuel				
321	0.4	0.2	0.1	
90	0.11	—	—	
.25	0.16	0.1	—	
250	0.31	0.2	0.1	
Number of Periods within stated percent of maximum				
90%		1	1	
50%		71	29	
NO_x Emissions (ppm)	(g/sec)	Annual NO_x Average ($\mu g/m^3$)		
Jet A Fuel				
55	0.11	2.7		
75	0.15	3.7		
100	0.21	4.9		
125	0.26	6.2		

— Concentrations less than 0.1

Table 6 - 3

Summary of Maximum Ground Level Concentrations

for Garret Airesearch Gas Turbine, Model GTC8S-90 (GPUT-90)

Cluster Arrangement (see Figure 6 -)

SO_2 Emissions (%S Fuel) Jet A Fuel	SO_2 Emissions (g/sec)	3-Hour SO_2 Average 20 Units ($\mu g/m^3$)	24-Hour SO_2 Average 20 Units ($\mu g/m^3$)
0.01	0.005	3.3	2.7
0.10	0.05	33.4	27.1
0.3	0.14	93.5	75.9
0.8	0.38	253.8	206.0

CO Emissions (ppm) Jet A Fuel	CO Emissions (g/sec)	1-Hour CO Average 20 Units (mg/m^3)	8-Hour CO Average 20 Units (mg/m^3)
321.0	0.4	0.3	0.3
90.0	0.11	0.1	0.1
125.0	0.16	0.1	0.1
250.0	0.31	0.2	0.2

— Concentrations less than 0.1

Table 6 - 4
Summary of Maximum Ground Level Concentrations for Solar Gas Turbine, Model Saturn

SO_2 Emissions ([illegible] Fuel)	(g/sec)	3-Hour SO_2 Average ($\mu g/m^3$)	24-Hour SO_2 Average ($\mu g/m^3$)	Annual SO_2 Average ($\mu g/m^3$)
Natural Gas				
0.0	0.0	0.0	0.0	0.0
Kerosene Fuel				
0.01	0.02	2.3	0.5	—
0.1	0.17	20.3	4.6	0.3
0.3	0.5	59.7	13.6	0.8
0.8	1.33	158.8	35.9	2.0
Number of Periods within stated percent of maximum				
90%		1	1	
50%		5	4	

CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m^3)	8-Hour CO Average (mg/m^3)
Natural Gas			
68.	0.27	—	—
90.	0.36	0.1	—
125.	0.49	0.1	—
250.	1.0	0.2	0.1
Kerosene Fuel			
163.	0.73	0.1	0.1
90.	0.41	0.1	—
125.	0.56	0.1	—
250.	1.12	0.2	0.1
Number of Periods within stated percent of maximum			
90%		8	1
50%		36	2

NO_x Emissions (ppm)	(g/sec)	Annual NO_x Average ($\mu g/m^3$)
Natural Gas		
51.	0.33	0.5
55.	0.36	0.5
75.	0.49	0.2
100.	0.66	1.0
Kerosene Fuel		
69.	0.51	0.8
75.	0.55	0.8
100	0.73	1.1
125	0.92	1.4

— Concentrations less than 0.1

Table 6 - 5
Summary of Maximum Ground Level Concentrations
for Solar Gas Turbine, Model Saturn
Cluster Arrangements (see Figure 6-)

SO_2 Emissions (%S Fuel)	SO_2 Emissions (g/sec)	3-Hour SO_2 Average 5 Units ($\mu g/m^3$)	3-Hour SO_2 Average 22 Units ($\mu g/m^3$)	24-Hour SO_2 Average 5 Units ($\mu g/m^3$)	24-Hour SO_2 Average 22 Units ($\mu g/m^3$)
Natural Gas					
0.0	0.0	0.0	0.0	0.0	0.0
Kerosene Fuel					
0.01	0.02	9.2	37.9	2.3	10.2
0.1	0.17	78.4	322.0	19.8	86.5
0.3	0.5	230.6	947.1	58.2	254.4
0.8	1.33	613.4	2519.2	154.9	676.7

CO Emissions (ppm)	CO Emissions (g/sec)	1-Hour CO Average 5 Units (mg/m^3)	1-Hour CO Average 22 Units (mg/m^3)	8-Hour CO Average 5 Units (mg/m^3)	8-Hour CO Average 22 Units (mg/m)
Natural Gas					
68.0	0.27	0.1	0.6	0.1	0.2
90.0	0.36	0.2	0.8	0.1	0.3
125.0	0.49	0.2	1.1	0.1	0.4
250.0	1.0	0.4	2.3	0.3	0.8
Kerosene Fuel					
163.0	0.73	0.3	1.7	0.2	0.6
90.0	0.41	0.2	0.9	0.1	0.3
125.0	0.56	0.2	1.30	0.2	0.5
250.0	1.12	0.5	2.6	0.3	0.9

— Concentrations less than 0.1

Table 6 - 6
Summary of Maximum Ground Level Concentrations
for Solar Gas Turbine, Model Centaur

SO_2 Emissions Fuel)	(g/sec)	3-Hour SO_2 Average ($\mu g/m^3$)	24-Hour SO_2 Average ($\mu g/m^3$)	Annual SO_2 Average ($\mu g/m^3$)
Natural Gas				
0.0	0.0	0.0	0.0	0.0
Kerosene Fuel				
0.01	0.045	1.7	0.3	—
0.1	0.45	17.0	3.0	0.2
0.3	1.34	50.9	9.1	0.5
0.8	3.58	135.9	24.4	1.4
Number of Periods within stated percent of maximum				
90%		1	2	
50%		11	11	
CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m^3)	8-Hour CO Average (mg/m^3)	
Natural Gas				
68.	0.73	—	—	
90.	0.97	0.1	—	
125.	1.35	0.1	—	
250.	2.7	0.2	0.1	
Kerosene Fuel				
126.	1.54	0.1	—	
90.	1.1	0.1	0.6	
125.	1.52	0.1	—	
250.	3.05	0.2	0.1	
Number of Periods within stated percent of maximum				
90%		3	1	
50%		31	12	
NO_x Emissions (ppm)	(g/sec)	Annual NO_x Average ($\mu g/m^3$)		
Natural Gas				
89	1.5	0.6		
55	0.9	0.4		
75	1.3	0.5		
100	1.8	0.7		
Kerosene Fuel				
86	1.7	0.7		
75	1.5	0.6		
100	2.0	0.8		
125	2.5	0.9		

— Concentrations less than 0.1

Table 6 - 7
Summary of Maximum Ground Level Concentrations
for Solar Gas Turbine, Model Centaur
Cluster Arrangement (see Figure 6 - 1)

SO_2 Emissions (%S Fuel)	(g/sec)	3-Hour SO_2 Average 5 Units (µg/m^3)	3-Hour SO_2 Average 22 Units (µg/m^3)	24-Hour SO_2 Average 5 Units (µg/m^3)	24-Hour SO_2 Average 22 Units (µg/m^3)
Natural Gas					
0.0	0.0				
Kerosene Fuel					
0.01	0.045	7.4	33.6	1.4	6.4
0.1	0.45	74.2	336.5	14.4	63.8
0.3	1.45	221.0	1002.0	43.0	190.0
0.8	3.58	590.4	2677.0	114.0	507.6

CO Emissions (ppm)	(g/sec)	1-Hour CO Average 5 Units (mg/m^3)	1-Hour CO Average 22 Units (mg/m^3)	8-Hour CO Average 5 Units (mg/m^3)	8-Hour CO Average 22 Units (mg/m^3)
Natural Gas					
68.0	0.73	0.1	0.4	0.1	0.1
90.0	0.97	0.1	0.6	0.1	0.2
125.0	1.35	0.2	0.8	0.1	0.3
250.0	2.7	0.3	1.6	0.2	0.5
Kerosene Fuel					
126.0	1.54	0.2	0.9	0.1	0.3
90.0	1.1	0.1	0.6	0.1	0.2
125.0	1.52	0.2	0.9	0.1	0.3
250.0	3.05	0.4	1.8	0.2	0.6

— Concentrations less than 0.1

Table 6 - 8
Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine,Model MS5001P

SO_2 Emissions %S Fuel)	(g/sec)	3-Hour SO_2 Average ($\mu g/m^3$)	24-Hour SO_2 Average ($\mu g/m^3$)	Annual SO_2 Average ($\mu g/m^3$)
DF-2 Fuel				
0.1	3.93	43.5	13.7	0.4
0.3	11.8	131.0	41.1	0.9
0.5	19.67	218.0	68.5	2.1
0.8	31.46	348.0	109.0	3.4
Doped DF-2 Fuel				
1.0	39.3	435.0	137.0	4.2
2.7	106.2	1175.0	369.0	11.2
0.3	11.8	131.0	41.1	0.9
0.8	31.46	384.0	109.0	3.4
Number of Periods within stated percent of maximum				
90%		3	3	
50%		27	16	
CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m^3)	8-Hour CO Average (mg/m^3)	
7.8	0.75	—	—	
90.0	8.65	—	0.1	
125.0	12.02	—	0.1	
250.0	24.04	—	0.2	
Doped DF-2 Fuel (.9% N)				
5.3	0.56	—	—	
90.0	8.65	0.1	0.1	
125.0	12.02	0.1	0.1	
250.0	24.04	0.3	0.2	
500.0	43.1	1.3	0.9	
Number of Periods within stated percent of maximum				
90%		6	2	
50%		53	13	
NO_x Emissions (ppm)	(g/sec)	Annual NO_x Average ($\mu g/m^3$)		
DF-2 Fuel				
147.0	23.5	2.5		
75.0	12.0	1.3		
100.0	16.0	1.7		
125.0	20.0	2.1		
Doped DF-2 Fuel (.9% N)				
245.0	41.9	4.4		
75.0	12.0	1.3		
00.0	16.0	1.7		
125.0	20.0	2.1		

Under Spinning Reserve Mode
for SO_2 emissions multiply by 0.33
For 3-Hour SO_2 concentrations multiply by 0.67
For 24-Hour SO_2 concentrations multiply by 0.79

— Concentrations less than 0.1

more than 8 units when fuel greater than 0.5% S is used under the regular mode of operation (Table 6-9). The SO_2 impact is less under the spinning reserve mode as a result of lower SO_2 emissions. The CO average concentrations are well below NAAQS even for the cluster arrangements. Under the spinning reserve mode of operation more occurrences of high concentrations are found due to the greater CO emissions and the lower plume rise.

The MS5002B unit is quite similar to the MS5001P, except it has a shorter stack and does not operate under the spinning reserve mode. The maximum ground level concentrations caused by the MS5002B are found to be 3 times greater than those caused by the MS5001P (Tables 6-10 and 6-11). However, at more distant receptors less difference in ground level concentrations are found.

The MS7001B units are the largest units considered in this analysis. Air quality impact for an individual simple cycle unit is estimated to be lower than NAAQS if less than 2% S fuel is used (Tables 6-12). Even with the reduced fuel rate taken into account under the spinning reserve mode, the maximum SO_2 impact is slightly greater than that found for the regular mode of operation. In order to meet the 3-hour SO_2 standard when a 16 unit cluster is in operation, 0.5% S fuel is necessary (Table 6-13). The unfired combined cycle MS7001B units are generally paired with their main stacks set beside each other. This configuration is assessed as two units with one main stack. The main stack diameter for this situation is

Table 6 - 9
Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine, Model MS5001P Cluster Arrangement (see Figure 6-1)

SO_2 Emissions			3-Hour SO_2 Average			24-Hour SO_2 Average		
(%S Fuel)	(g/sec)		8 Units RM ($\mu g/m^3$)	16 Units RM ($\mu g/m^3$)	16 Units SR ($\mu g/m^3$)	8 Units RM ($\mu g/m^3$)	16 Units RM ($\mu g/m^3$)	16 Units SR ($\mu g/m^3$)
DF-2 Fuel								
(RM,SR)	(RM)	(SR)						
0.1	3.93	1.31	243.8	245.0	176.6	75.4	99.1	81.0
0.3	11.8	3.93	732.0	735.6	530.1	226.4	297.7	243.2
0.5	19.7	6.57	1222.0	1228.0	885.0	378.0	497.0	406.0
0.8	31.5	10.5	1954.0	1963.6	1415.1	604.4	794.7	649.2
Doped DF-2 Fuel (.9% N)								
1.0	39.3	13.1	2446.6	2456.6	1771.8	756.0	926.9	812.8
2.7	106.0	35.3	6599.0	6626.0	4779.0	2039.0	2500.0	2192.4
0.3	11.8	3.93	734.6	737.6	532.0	227.0	278.3	244.1
0.8	31.5	10.5	1961.0	1969.0	1420.2	605.9	742.9	651.5
CO Emissions			1-Hour CO Average			8-Hour CO Average		
(ppm)	(g/sec)		8 Units RM (mg/m^3)	16 Units RM (mg/m^3)	16 Units SR (mg/m^3)	8 Units RM (mg/m^3)	16 Units RM (mg/m^3)	16 Units SR (mg/m^3)
DF-2 Fuel								
(RM)	(RM)							
7.8	0.75		0.1	0.1		—	—	
90.0	8.65		0.8	0.8		.4	.5	
125.0	12.0		1.0	1.0		.6	.7	
250.0	24.0		2.1	2.1		1.2	1.4	
Doped DF-2 Fuel (.9%N)								
(RM)	(RM)							
5.3	0.56		—	—		—	—	
90.0	8.56		0.8	0.8		.4	.5	
125.0	12.0		1.0	1.0		.6	.7	
250.0	24.0		2.1	2.1		1.2	1.4	
(Spinning Reserve)								
500.0	43.1				7.8			5.6

— Concentrations less than 0.1

Table 6 - 10
Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine, Model MS5002B

SO$_2$ Emissions (%S Fuel)	(g/sec)	3-Hour SO$_2$ Average (µg/m^3)	24-Hour SO$_2$ Average (µg/m^3)	Annual SO$_2$ Average (µg/m^3)
DF-2 Fuel				
0.1	3.93	139.0	47.3	2.4
0.3	11.8	418.0	142.0	8.2
0.5	19.7	697.0	237.0	11.9
0.8	31.5	1115.0	379.0	19.0
Doped DF-2 Fuel (.9% N)				
1.0	39.3	1393.0	473.0	23.8
2.7	10.6	3764.0	1278.0	64.7
0.3	11.8	418.0	142.0	8.2
0.8	31.5	1115.0	379.0	19.0
Number of Periods within stated percent of maximum				
90%		2	2	
50%		21	14	
CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m^3)	8-Hour CO Average (mg/m^3)	
DF-2 Fuel				
7.8	0.75	-	-	
90	8.65	0.4	0.3	
125	12.0	0.6	0.4	
250	24.0	1.2	0.7	
Doped DF-2 Fuel				
5.3	0.56	-	-	
90	8.65	0.4	0.3	
125	12.0	0.6	0.4	
250	24.0	1.2	0.7	
Number of Periods within stated percent of maximum				
90%		4	1	
50%		40	6	
NO$_x$ Emissions (ppm)	(g/sec)	Annual NO$_x$ Average (µg/m^3)		
DF-2 Fuel				
147	23.5	14.3		
75	12.0	7.3		
100	16.0	9.7		
125	20.0	12.2		
Doped DF-2 Fuel				
245	41.9	25.5		
75	12.0	7.3		
100	16.0	9.7		
125	20.0	12.2		

— Concentrations less than 0.1

Table 6 - 11
Summary of Maximum Ground Level Concentrations
for General Electric Gas Turbine, Model MS5002B
Cluster Arrangement (see Figure 6 - 1)

SO$_2$ Emissions		3-Hour SO$_2$ Average			24-Hour SO$_2$ Average		
(%S Fuel)	(g/sec)	2 Units (µg/m^3)	4 Units (µg/m^3)	8 Units (µg/m^3)	2 Units (µg/m^3)	4 Units (µg/m^3)	8 Units (µg/m^3)
DF-2 Fuel							
0.1	3.93	256.3	498.9	723.8	86.8	152.4	239.4
0.3	11.8	769.7	1347.7	2173.1	260.6	457.6	718.8
0.5	19.7	1285.0	2250.0	3628.0	435.0	764.0	1200.0
0.8	31.5	2054.7	3597.7	5801.1	695.6	1221.6	1918.8
Doped DF-2 Fuel (.9% N)							
1.0	39.3	2563.5	4488.6	7237.6	867.8	1524.1	2393.9
2.7	106.0	6914.2	12106.6	19521.2	2340.6	4110.9	6456.9
0.3	11.8	769.7	1347.7	2173.1	260.6	457.6	718.8
0.8	31.5	2054.7	3597.7	5801.1	695.6	1221.6	1918.8

CO Emissions		1-Hour CO Average			8-Hour CO Average		
(ppm)	(g/sec)	2 Units (mg/m^3)	4 Units (mg/m^3)	8 Units (mg/m^3)	2 Units (mg/m^3)	4 Units (mg/m^3)	8 Units (mg/m^3)
DF-2 Fuel							
7.8	0.75	0.1	0.1	0.2	—	0.1	0.1
90.0	8.65	0.8	1.3	2.1	0.5	0.8	1.3
125.0	12.0	1.0	1.3	2.9	0.6	1.1	1.8
250.0	24.0	2.1	3.7	5.9	1.3	2.3	3.7
Doped DF-2 Fuel (.9% N)							
5.3	0.56	—	0.1	0.2	—	0.1	0.1
90.0	8.65	0.8	1.3	2.1	0.5	0.8	1.3
125.0	12.0	1.0	1.8	2.9	0.6	1.1	1.8
250.0	24.0	2.1	3.7	5.9	1.3	2.3	3.7

— Concentrations less than 0.1

Table 6 -12
Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine, Model MS7001B (Simplc Cycle)

SO_2 Emissions (%S Fuel)	(g/sec)	3-Hour SO_2 Average (μg/m³)	24-Hour SO_2 Average (μg/m³)	Annual SO_2 Avera (μg/m³)
Natural Gas				
0.0	0.0	0.0	0.0	0.0
DF-2 Fuel				
0.3	26.5	212.0	61.0	0.3
1.0	88.2	709.0	202.0	1.1
2.7	238.0	1906.0	548.0	2.9
0.1	8.8	71.0	20.0	0.1
0.8	70.6	565.0	162.0	0.9
Number of Periods within stated percent of maximum				
90%		1	1	
50%		6	1	

CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m³)	8-Hour CO Average (mg/m³)
Natural Gas			
2.3	0.39	—	—
90.0	15.3	0.2	0.1
125.0	21.2	0.3	0.1
250.0	42.4	0.5	0.3
DF-2 Fuel			
90.0	15.7	0.2	0.1
125.0	21.8	0.3	0.1
250.0	43.6	0.5	0.3
(Spinning Reserve)			
500	87.2	2.4	1.5
Number of Periods within stated percent of maximum			
90%		4	1
50%		27	1

NO_x Emissions (ppm)	(g/sec)	Annual NO_x Average (μg/m³)
Natural Gas		
119.0	33.4	0.4
55.0	15.4	0.2
75.0	21.1	0.2
100.0	28.1	0.3
DF-2 Fuel		
163.0	46.7	0.5
75.0	21.5	0.2
100.0	28.7	0.3
125.0	35.8	0.4

Under Spinning Reserve Mode
for SO_2 emissions multiply by 0.3
for 3-Hour SO_2 concentrations multiply by 0.83
for 24-Hour SO_2 concentrations multiply by 0.98

— Concentrations less than 0.1

Table 6 - 13

Summary of Maximum Ground Level Concentrations

for General Electric Gas Turbine, Model MS7001B (Simple Cycle)

Cluster Arrangement (see Figure 6-1)

Regular Mode (RM)

Spinning Reserve Mode (SR)

SO_2 Emissions (%S Fuel)	(g/sec)		3-Hour SO_2 Average 8 Units RM ($\mu g/m^3$)	16 Units RM ($\mu g/m^3$)	16 Units SR ($\mu g/m^3$)	24-Hour SO_2 Average 8 Units RM ($\mu g/m^3$)	16 Units RM ($\mu g/m^3$)	16 Units SR ($\mu g/m^3$)
Natural Gas								
0.0	0.0		0.0	0.0	0.0	0.0	0.0	0.0
DF-2 Fuel								
(RM/SR)	(RM)	(SR)						
0.3	26.5	8.8	636.8	777.4	786.2	65.1	212.9	246.2
1.0	88.2	29.4	2119.4	7587.4	2616.7	216.8	708.6	819.4
2.7	238.0	70.3	5719.0	6982.0	7061.0	585.0	1912.0	2211.0
0.1	8.82	2.9	211.9	258.7	261.7	21.7	70.9	81.9
0.8	70.6	23.5	1696.5	2071.1	2094.6	173.5	567.2	655.9

CO Emissions (ppm)	(g/sec)	1-Hour CO Average 8 Units RM (mg/m^3)	16 Units RM (mg/m^3)	16 Units SR (mg/m^3)	8-Hour CO Average 8 Units RM (mg/m^3)	16 Units RM (mg/m^3)	16 Units SR (mg/m^3)
Natural Gas							
2.3	0.39	—	—		—	—	
90.0	15.3	0.6	0.7		0.3	0.3	
125.0	21.2	0.8	0.9		0.4	0.4	
250.0	42.4	1.6	1.9		0.8	0.9	
DF-2 Fuel							
90.0	15.7	0.6	0.7		0.3	0.3	
125.0	21.8	0.8	1.0		0.4	0.5	
250.0	43.6	1.6	1.9		0.8	0.9	
(Spinning Reserve)							
500.0	84.8			10.8			6.3

— Concentrations less than 0.1

based on an equivalent double-unit circle. The by-pass stacks are treated individually for this configuration.

The MS7001B unfired combined cycle unit is assessed as a pair of machines. The reported concentrations are, however, given as those which occur per machine (Table 6-14). Since the stack height is significantly higher and the mass flow rate is very large for the main stack, higher plume rise and thus lower concentrations occur near the source than for the other General Electric units. The maximum 1-hour and 3-hour average concentrations are found to occur at 0.5 km. It should be noted that concentrations occurring closer to the source are primarily a result of the much smaller by-pass stack emissions. As long as 2.7% S or greater fuel is not used, impact of even a 16 unit cluster is estimated to be less than all NAAQS (Table 6-15).

The highest ground level concentrations of all units examined are found for the MS7001B (regenerative) unit. Less than 0.8% S fuel is necessary in order not to exceed short-term SO_2 air quality standards at 0.1 km for an individual unit (Table 6-16) under the regular mode of operation. Under the spinning reserve mode of operation, maximum SO_2 concentrations are found to be slightly lower. The only situation in all the analyses of this report that CO concentration standards are estimated to be exceeded is for the 8-hour average when greater than 8 units are operating under the spinning reserve mode (Table 6-17). In order to meet the standards, it is estimated that 20% control is needed for 8 units and 50% control is needed for 16 units. In order not to exceed the SO_2 short-term standards at 0.1 km

Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine, Model MS7001B(Unfired Combined Cycle)

SO_2 Emissions Fuel	(g/sec)	3-Hour SO_2 Average ($\mu g/m^3$)	24-Hour SO_2 Average ($\mu g/m^3$)	Annual SO_2 Average ($\mu g/m^3$)
Natural Gas				
0.0	0.0	0.0	0.0	0.0
DF-2 Fuel				
0.3	26.5	36.3	17.5	2.1
1.0	88.2	121.0	58.2	7.1
2.7	238.0	326.0	157.0	19.1
0.1	8.8	12.1	5.8	0.7
0.8	70.6	96.7	46.6	5.7
umber of Periods ithin stated percent f maximum				
90%		1	1	
50%		5	41	

CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m^3)	8-Hour CO Average (mg/m^3)
Natural Gas			
2.3	0.39	—	—
90.0	15.3	—	—
125.0	21.2	—	—
250.0	42.4	0.1	—
DF-2 Fuel			
90.0	15.7	—	—
125.0	21.8	—	—
	43.6	0.1	—
lumber of Periods vithin stated percent of maximum			
90%		3	2
50%		23	41

NO_x Emissions (ppm)	(g/sec)	Annual NO_x Average ($\mu g/m^3$)
Natural Gas		
119.0	33.4	0.4
55.0	15.4	0.2
75.0	21.1	0.3
100.0	28.1	0.3
DF-2 Fuel		
163.0	46.7	0.5
75.0	21.5	0.3
100.0	28.7	0.3
125.0	35.8	0.4

— Concentrations less than 0.1

Table 6 - 15

Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine, Model MS7001B (Unfired Combined Cycle) Cluster Arrangement (See Figure 6 -1)

Regular Mode (RM)
Spinning Reserve Mode (SR)

SO_2 Emissions (%S Fuel)	SO_2 Emissions (g/sec)	3-Hour SO_2 Average 16 Units (μg/m^3)	24-Hour SO_2 Average 16 Units (μg/m^3)
Natural Gas			
0.0	0.0	0.0	0.0
DF-2 Fuel			
0.3	26.5	315.3	128.4
1.0	88.2	1049.5	427.3
2.7	238.0	2832.0	1153.0
0.1	8.82	105.0	42.7
0.8	70.6	840.1	342.0

CO Emissions (ppm)	CO Emissions (g/sec)	1-Hour CO Average 16 Units (mg/m^3)	8-Hour CO Average 16 Units (mg/m^3)
Natural Gas			
2.3	0.39	—	—
90.0	15.3	0.4	0.1
125.0	21.2	0.5	0.2
250.0	42.4	0.7	0.3
DF-2 Fuel			
90.0	15.7	0.4	0.1
125.0	21.8	0.5	0.2
250.0	43.6	0.7	0.3

— Concentrations less than 0.1

Table 6 -16
Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine, Model MS7001B (Regenerative Cycle)

SO_2 Emissions Fuel)	(g/sec)	3-Hour SO_2 Average ($\mu g/m^3$)	24-Hour SO_2 Average ($\mu g/m^3$)	Annual SO_2 Average ($\mu g/m^3$)
DF-2 Fuel				
0.1	6.4	160.0	55.0	2.5
0.3	19.3	180.0	166.0	7.6
0.5	22.2	790.0	277.0	8.8
1.0	64.4	1580.0	550.0	25.5
2.7	174.0	4280.0	1496.0	68.9
0.8	51.5	1270.0	443.0	20.4
Number of Periods within stated percent of maximum				
90%		3	2	
50%		21	14	
CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m^3)	8-Hour CO Average (mg/m^3)	
DF-2Fuel				
5.0	0.97	—	—	
90.0	17.5	0.6	0.4	
125.0	21.2	0.7	0.5	
250.0	42.4	1.4	0.9	
(Spinning Reserve)				
500.0	84.8	5.3	3.5	
Number of Periods within stated percent of maximum				
90%		4	1	
50%		41	7	
NO_x Emissions (ppm)	(g/sec)	Annual NO_x Average ($\mu g/m^3$)		
DF-2 Fuel				
373.0	118.0	49.9		
75.0	23.7	10.0		
100.0	31.6	13.4		
125.0	39.5	16.7		

Under Spinning Reserve Mode
for SO_2 emissions multiply by 0.33
for 3-Hour SO_2 concentrations multiply by 0.63
for 24-Hour SO_2 concentrations multiply by 0.95

— Concentrations less than 0.1

Table 6 - 17

Summary of Maximum Ground Level Concentrations
for General Electric Gas Turbine, Model MS7001B (Regenerative Cycle)
Cluster Arrangement (see Figure 6 -1)

Regular Mode (RM)
Spinning Reserve Mode (SR)

SO_2 Emissions (%S Fuel) (RM,SR)	(g/sec) DF-2 Fuel (RM)	(SR)	3-Hour SO_2 Average 8 Units RM ($\mu g/m^3$)	16 Units RM ($\mu g/m^3$)	16 Units SR ($\mu g/m^3$)	24-Hour SO_2 Average 8 Units RM ($\mu g/m^3$)	16 Units RM ($\mu g/m^3$)	16 Units SR ($\mu g/m^3$)
0.1	6.44	2.185	587.0	819.4	558.8	161.6	279.0	229.7
0.3	19.3	6.4	1759.2	1759.2	1674.7	484.3	836.1	688.5
0.5	32.2	10.7	2935.0	4097.2	2794.0	807.9	1395.0	1148.7
1.0	64.4	21.5	5870.0	8194.3	5588.0	1615.9	2789.9	2297.3
2.7	174.0	58.0	15,960.0	22,140.0	15,098.0	4366.0	7538.0	6207.0
0.8	51.5	17.2	4694.2	4694.2	4468.7	1292.2	2231.1	1837.1

CO Emissions (ppm)	(g/sec) DF-2 Fuel	1-Hour CO Average 8 Units RM (mg/m^3)	16 Units RM (mg/m^3)	16 Units SR (mg/m^3)	8-Hour CO Average 8 Units RM (mg/m^3)	16 Units RM (mg/m^3)	16 Units SR (mg/m^3)
5.0	0.97	0.1	0.2		0.1	0.1	
90.0	17.5	2.0	2.8		1.2	1.7	
125.0	21.2	2.4	3.4		1.5	2.2	
250.0	42.4	4.8	6.8		3.0	4.2	
(Spinning Reserve)							
500.0	84.8			26.1			18.2

— Concentrations less than 0.1

when a group of 16 units are operating (regular mode) in a cluster, it is estimated that less than 0.2% S fuel is necessary (Table 6-17). Concentrations found at 0.2 km and 0.5 km are one-half and one-fourth respectively, of those found at 0.1 km for the operating 16 unit cluster. Thus, if the air quality region of concern is beyond these further distances, less stringent control is needed.

The results of the analysis (Tables 6-18 and 6-19) of the MS7001C are nearly identical to those for the MS7001B (Simple Cycle). The discussion for the MS7001B (Simple Cycle) is thus applicable.

6.1.5 Impact of Variations in Meteorological Conditions

In general, the maximum estimated concentrations occur during the windiest hours for most turbines considered. Should windier situations exist, even higher concentrations may occur. In addition, concentrations would be higher if situations exist when wind direction variability is smaller.

To provide an evaluation of how windier conditions effect maximum ground level concentrations, further analyses have been conducted for the MS7001B (Regenerative Cycle). This unit is selected since it has the worst air quality impact. Tables 6-20 to 6-22 contain the velocity increase evaluation for the worst case meteorological conditions with only the wind speed modified. Under the normal mode of operation, a significant increase in the maximum concentrations that occur at 0.1 km is found for the 1.5 times and 2.0 times wind speed cases. Wind speeds

Table 6 - 18
Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine, Model MS7001C

SO_2 Emissions (%S Fuel)	(g/sec)	3-Hour SO_2 Average ($\mu g/m^3$)	24-Hour SO_2 Average ($\mu g/m^3$)	Annual SO_2 Average ($\mu g/m^3$)
DF-2 Fuel				
0.1	9.9	62.6	10.7	—
0.3	27.8	175.0	29.6	0.1
0.5	49.6	313.0	53.1	0.2
1.0	92.3	583.0	98.8	0.3
2.7	268.0	1691.0	287.0	0.9
0.8	79.4	501.0	85.0	0.3
Number of Periods within stated percent of maximum				
90%		1	1	
50%		2	3	

CO Emissions (ppm)	(g/sec)	1-Hour CO Average (mg/m^3)	8-Hour CO Average (mg/m^3)
DF-2 Fuel			
2.1	0.51	—	—
90.0	21.9	0.2	0.1
125.0	30.4	0.3	0.1
250.0	60.7	0.6	0.2
(Spinning Reserve)			
500.0	121.0	2.7	1.7
Number of Periods within stated percent of maximum			
90%		4	1
50%		17	3

NO_x Emissions (ppm)	(g/sec)	Annual NO_x Average ($\mu g/m^3$)
DF-2 Fuel		
164.0	66.9	0.2
75.0	30.6	0.1
100.0	40.8	0.1
125.0	51.0	0.2

Under Spinning Reserve Mode
for SO_2 emissions multiply by 0.33
for 3-Hour SO_2 concentrations multiply by 0.6
for 8-Hour SO_2 concentrations multiply by 1.6

— Concentrations less than 0.1

Table 6 -19

Summary of Maximum Ground Level Concentrations for General Electric Gas Turbine, Model MS7001C Cluster Arrangement (see Figure 6 -1)

Regular Mode (RM)
Spinning Reserve Mode (SR)

SO_2 Emissions (%S Fuel) (RM,SR)	(g/sec) DF-2 Fuel (RM)	(g/sec) DF-2 Fuel (SR)	3-Hour SO_2 Average 8 Units RM (μg/m^3)	3-Hour SO_2 Average 16 Units RM (μg/m^3)	3-Hour SO_2 Average 16 Units SR (μg/m^3)	24-Hour SO_2 Average 8 Units RM (μg/m^3)	24-Hour SO_2 Average 16 Units RM (μg/m^3)	24-Hour SO_2 Average 16 Units SR (μg/m^3)
0.1	9.93	3.31	180.3	217.4	227.6	30.3	36.8	68.5
0.3	27.8	9.27	504.8	608.7	637.3	85.0	103.0	191.9
0.5	49.6	16.5	900.6	1086.0	1137.1	151.6	183.8	342.4
1.0	92.3	30.8	1675.9	2021.0	2116.0	282.1	342.0	637.1
2.7	268.0	89.3	4866.0	5868.0	6144.0	819.0	993.0	1850.0
0.8	79.4	26.5	1441.6	1738.5	1820.3	242.6	294.2	548.1

CO Emissions (ppm)	(g/sec) DF-2 Fuel	1-Hour CO Average 8 Units RM (mg/m^3)	1-Hour CO Average 16 Units RM (mg/m^3)	1-Hour CO Average 16 Units SR (mg/m^3)	8-Hour CO Average 8 Units RM (mg/m^3)	8-Hour CO Average 16 Units RM (mg/m^3)	8-Hour CO Average 16 Units SR (mg/m^3)
2.1	0.51	—	—		—	—	
90.0	21.9	0.4	0.5		0.1	0.2	
125.0	30.4	0.6	0.7		0.2	0.2	
250.0	60.7	1.2	1.4		0.4	0.5	
	(Spinning Reserve)						
500.0	121.4			12.3			6.8

— Concentrations less than 0.1

Table 6 - 20

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Regenerative Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964
Evaluation of Velocity Increase

(PTMTP Model Estimates)

Downwind Distance (km)	3-Hour SO_2 Average 1.0% S Q=64.4g/sec 1.5 Times ($\mu g/m^3$)	No Increase ($\mu g/m^3$)	2.0 Times ($\mu g/m^3$)	24-Hour SO_2 Average 1.0% S Q=64.4g/sec 1.5 Times ($\mu g/m^3$)	No Increase ($\mu g/m^3$)	2.0 Times ($\mu g/m^3$)
.1	1960.0	1354.0	2034.0	827.0	466.0	959.0
.2	868.0	516.0	945.0	345.0	166.0	475.0
.5	283.0	160.0	311.0	109.0	50.0	137.0
2.0	58.7	48.9	54.7	25.4	17.4	26.6
20.0	2.9	4.2	2.2	1.7	2.3	1.3

Downwind Distance (km)	1-Hour CO Average 250 ppm Q=42.4g/sec 1.5 Times (mg/m^3)	No Increase (mg/m^3)	2.0 Times (mg/m^3)	8-Hour CO Average 250 ppm Q=42.4g/sec 1.5 Times (mg/m^3)	No Increase (mg/m^3)	2.0 Times (mg/m^3)
.1	1.6	1.2	1.0	1.2	0.8	1.3
.2	0.8	0.5	0.8	0.5	0.3	0.6
.5	0.2	0.2	0.3	0.2	0.1	0.2
2.0	0.1	—	—	—	—	—
20.0	—	—	—	—	—	—

— Concentrations less than 0.1

Table 6 - 21

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Regenerative Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964
16 Unit Cluster (see Figure 6-6)
Evaluation of Velocity Increase

(PTMTP Model Estimates)

Downwind Distance	3-Hour SO_2 Average 1.0% S Q=64.4 g/sec			24-Hour SO_2 Average 1.0% S Q=64.4 g/sec		
(km)	1.5 Times ($\mu g/m^3$)	No Increase ($\mu g/m^3$)	2.0 Times ($\mu g/m^3$)	1.5 Times ($\mu g/m^3$)	No Increase ($\mu g/m^3$	2.0 Times ($\mu g/m^3$)
.1	13130.0	8200.0	14060.0	5804.0	2792.0	7202.0
.2	8263.0	4779.0	9032.0	3363.0	1525.0	4254.0
.5	3927.0	2020.0	3808.0	1381.0	637.0	1725.0
2.0	896.0	781.0	823.0	387.0	284.0	408.0
20.0	45.9	66.1	34.9	26.4	36.8	20.3

	1-Hour CO Average 250 ppm Q=42.4 g/sec			8-Hour CO Average 250 ppm Q=42.4 g/sec		
	1.5 Times (mg/m^3)	No Increase (mg/m^3)	2.0 Times (mg/m^3)	1.5 Times (mg/m^3)	No Increase (mg/m^3)	2.0 Times (mg/m^3)
.1	10.1	6.8	10.5	7.3	4.2	7.9
.2	6.8	4.3	7.2	4.7	2.5	5.3
.5	3.1	1.9	3.2	2.1	1.1	2.3
2.0	0.8	0.7	0.7	0.6	0.5	0.5
20.0	—	0.1	—	0.3	—	—

— Concentrations less than 0.1

Table 6 - 22

GENERAL ELECTRIC

MS7001B; DF-2 FUEL
Regenerative Cycle
(Spinning Reserve Mode)
Maximum Ground Level Concentration; Dallas, Texas, 1964
Evaluation of Velocity Increase

(PTMTP Model Estimates)

Downwind Distance	3-Hour SO_2 Average 1.0% S^2 Q=21.5 g/sec			24-Hour SO_2 Average 1.0% S^2 Q=21.5g/sec		
(km)	1.5 Times ($\mu g/m^3$)	No Increase ($\mu g/m^3$)	2.0 Times ($\mu g/m^3$)	1.5 Times ($\mu g/m^3$)	No Increase ($\mu g/m^3$)	2.0 Times ($\mu g/m^3$)
.1	922.0	849.0	852.0	438.0	338.0	431.0
.2	429.0	367.0	406.0	195.0	137.0	203.0
.5	131.0	119.0	132.0	62.5	42.3	65.4
2.0	27.5	33.4	22.4	14.4	15.3	17.4
20.0	0.9	1.5	0.8	0.6	0.8	0.4

	1-Hour CO Average 500 ppm Q=84.8g/sec			8-Hour SO_2 Average 500 ppm Q=84.8g/sec		
	1.5 Times (mg/m^3)	No Increase (mg/m^3)	2.0 Times (mg/m^3)	1.5 Times (mg/m^3	No Increase (mg/m^3)	2.0 Times (mg/m^3)
.1	4.4	4.3	4.0	3.4	3.0	3.2
.2	2.0	1.9	2.0	1.6	1.2	1.5
.5	0.7	0.6	0.7	0.5	0.5	0.5
2.0	0.1	0.2	0.1	0.1	0.1	0.9
20.0	—	—	—	—	—	—

— Concentrations less than 0.1

up to 25 m/s are considered in the 2.0 times cases. It is important though to note that the set of worst-case meteorological conditions selected as input to the modified PTMTP Model are those which resulted in the maximum value among all the receptors considered. The concentration calculations reported for the receptors downwind from the maximum (0.1 km receptor) do not represent the potential impact, since the maximum ground level concentrations for each receptor will occur in most cases under different meteorological conditions. The worst potential impact of a cluster arrangement at 2.0 km and greater distances is best determined by assuming a simple multiplication of the number of units times the maximum concentration calculated from the Single Source Model analysis of the individual unit. Under the spinning reserve mode of operation, the increase in wind speed has a much smaller impact and in some cases results in a decrease in maximum concentrations at 0.1 km. Note that for both modes of operation the concentrations at the more distant receptor decreases as the wind speed increases since plume rise has a relatively smaller change and the dilution effects predominate. These increased wind speeds are possible occurrences; however, those at the upper end of the range considered are not expected to occur very frequently.

Under high winds, concentrations at about 0.1 km for the MS7001B (regenerative) are estimated to exceed the NAAQS, even if less than .3% S fuel is used. The adverse impact from stationary gas turbines is mainly a result of the low profile that the effluent stacks have. A modest increase in the stack heights to about 2.0 times the height of the highest surrounding structures would have a very significant impact on air quality. Tables 6-23 and 6-24 summarize the impact of stack height increase. The MS7001B (Regenerative Cycle) is again used as the demonstration case. Significant decreases in concentration, especially at the closer receptors, are estimated to occur.

6.1.6 Summary

Nine gas turbine units are analyzed on an individual basis and in selected cluster arrangements. Only one situation is estimated to cause concentrations which exceed the CO NAAQS. This situation requires that greater than 8 units be operating in the spinning reserve mode. Many situations are identified where the short-term SO_2 standards could be exceeded. Annual SO_2 and NO_2 average concentration estimates are not found to exceed air quality standards for individual units. Although not specifically analyzed, annual average concentrations caused by cluster arrangements are not expected to be large enough to exceed standards. Only the General Electric MS5002B and MS7001B (Regenerative Cycle) gas turbines indicate any potential for exceeding NAAQS. If SO_2 emissions are controlled to meet short-term NAAQS, the annual SO_2 concentrations will definitely be below the standards. For the short-term averages the maximum impact of 16 units is approximately 5 times that occurring for an individual unit (Tables 6-10, 6-11, and 6-16, 6-17). If on an annual basis the factor of 5 multiplication holds, the maximum ground level concentrations from 16 uncontrolled units could potentially exceed the NAAQS for NO_2. It is most likely

Table 6 - 23

(Single Source Model Estimates)

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Regenerative Cycle
Maximum Ground Level Concentration; Dallas Texas, 1964
Evaluation of Stack Height Increase

Downwind Distance	3-Hour SO_2 Average 1.0% S Q=64.4g/sec		(No Increase)	24-Hour SO_2 Average 1.0% S Q=64.4g/sec		(No Increase)	Annual SO_2 Average 1.0% S Q=64.4g/sec		(No Increa
	$H_s/H_b=1.5$	$H_s/H_b=2.0$	$H_s/H_b=1.1$	$H_s/H_b=1.5$	$H_s/H_b=2.1$	$H_s/H_b=1.1$	$H_s/H_b=1.5$	$H_s/H_b=2.0$	$H_s/H_b=$
(km)	(μg/m^3)	(μg/m^3)	(μg/m^3)	(μg/m^3)	(μg/m^3)	(μg/m^3)	(μg/m^3)	(μg/m^3)	(μg/m
.1	869.0	2.7	1581.0	283.0	0.3	550.0	5.6	—	26.5
.2	382.0	71.0	561.0	120.0	9.1	286.0	2.3	0.1	6.8
.5	144.0	121.0	269.0	45.0	19.9	53.9	1.3	0.7	2.0
2.0	50.8	49.4	51.4	18.6	17.2	18.7	1.8	1.8	1.9
20.0	39.6	40.3	50.5	7.9	7.0	8.6	0.9	0.9	0.9

	1-Hour CO Average 250 ppm Q=42.4g/sec		(No Increase)	8-Hour CO Average 250 ppm Q=42.4g/sec		(No Increase)
	$H_s/H_b=1.5$	$H_s/H_b=2.0$	$H_s/H_b=1.1$	$H_s/H_b=1.5$	$H_s/H_b=2.0$	$H_s/H_b=1.0$
	(mg/m^3)	(mg/m^3)	(mg/m^3)	(mg/m^3)	(mg/m^3)	(mg/m^3)
.1	0.8	—	1.4	0.5	—	0.9
.2	0.4	0.1	0.6	0.2	—	0.3
.5	0.2	0.1	0.2	0.1	—	0.1
2.0	—	0.1	0.1	—	—	0.3
20.0	0.1	0.1	0.1	—	—	—

— Concentrations are less than .1

(Single Source Model Estimates)

Table 6 - 24

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Regenerative Cycle
(Spinning Reserve Mode)
Maximum Ground Level Concentration; Dallas, Texas, 1964
Evaluation of Stack Height Increase

Downwind Distance	3-Hour SO_2	Average 1.0% S Q=21.5g/sec	(No Increase)	24-Hour SO_2	Average 1.0% S Q=21.5g/sec	(No Increase)	Annual SO_2	Average 1.0% S Q=21.5g/sec	(No Increase
	H_s/H_b=1.5	H_s/H_b=2.0	H_s/H_b=1.1	H_s/H_b=1.5	H_s/H_b=2.0	H_s/H_b=1.1	H_s/H_b=1.5	H_s/H_b=2.0	H_s/H_b=1.
(km)	(μg/m³)	(μg/m³)	(μg/m³)	(μg/m³)	(μg/m³)	(μg/m³)	(μg/m³)	(μg/m³)	(μg/m³)
.1	555.0	238.0	1014.0	288.0	69.3	543.0	18.7	0.3	42.0
.2	286.0	153.0	402.0	121.0	44.6	181.0	7.9	0.4	13.3
.5	113.0	78.7	128.0	45.5	25.0	52.0	2.0	1.2	4.0
2.0	46.5	41.3	53.0	29.8	26.5	31.0	3.1	2.8	3.3
20.0	24.3	21.4	24.0	4.4	3.4	5.0	0.4	0.4	0.4

	1-Hour CO	Average 500 ppm Q=84.8g/sec	(No Increase)	8-Hour CO	Average 500 ppm Q=84.8g/sec	(No Increase)
	H_s/H_b=1.5	H_s/H_b=2.0	H_s/H_b=1.1	H_s/H_b=1.5	H_s/H_b=2.0	H_s/H_b=1.1
	(mg/m³)	(mg/m³)	(mg/m³)	(mg/m³)	(mg/m³)	(mg/m³)
.1	2.9	1.2	5.3	1.9	0.7	3.4
.2	1.4	0.8	2.1	0.0	1.0	1.4
.5	0.6	0.4	0.7	0.4	0.4	0.4
2.0	0.2	0.2	0.3	0.1	0.1	0.2
20.0	0.2	0.2	0.3	—	—	0.1

— Concentrations are less than .1

that for an annual average the maximum impact from 16 units will be somewhat less.

An examination of meteorological conditions associated with maximum concentrations indicates that, in most cases, higher concentrations can occur in areas with stronger and steadier winds. Also, it is found that concentrations can be considerably reduced by an increase of stack height.

Discussion in the above sections reports the concentration estimates made at the receptor of maximum concentration (.1 km for most cases). The concentration estimates at the additional receptors are included in Appendix E (Tables E-1 through E-26). Because of the large volume of data, the results in Appendix E are reported for only one set of emission levels. For the analyses of individual turbines and the cluster arrangements having all turbines operating in the same mode, concentration estimates for other emission rates change proportionately.

6.2 WATER POLLUTION IMPACT

Promulgation of Federal standards of performance for stationary gas turbines will have little impact on water pollution. Currently, the only potential source of water pollution is from the water treatment system for the water used to control NO_x emissions. Since manufacturers of gas turbines predict that they will be able to meet NO_x emission limits within five years using dry control techniques (combustor modifications), there will be no impact on water pollution after dry techniques are applied.

Relative consumption rates of water for alternative means of generating power are shown in Table 4.1 and discussed in section 4.5.1.1.1 which show that the consumptive water use of a conventional steam boiler is over 20 times the water which might be used to control NO_x emissions from a comparable gas turbine. The boiler water and the water used for injection into the turbine both must be of high quality and the same water treatment processes are used. The boiler facility also uses water for other purposes such as cooling, blowdown, fly-ash sluicing, wet scrubbers, etc. Therefore, the potential for water pollution from a boiler is many times greater than that for an equivalent turbine.

A typical water purification system is shown in Figure 4.15. This water treatment system for five 28 MW gas turbines operating 10 hours per day treats 125,000 gal/day of water and rejects about 25,000 gallons per day of waste water. The 5,000 gallons used as backwash and rinse water for the mixed bed demineralizer is treated in a neutralizing tank for pH adjustment prior to discharge to the municipal sewer, return to a river, or evaporation in a pond. The 20,000 gallons per day of reject water from the reverse osmosis system is sewered directly, returned to the river or evaporated in ponds. The quality of the waste water prior to sewering or return to the river is essentially the same as the influent water except that the concentration of total dissolved solids in the waste stream is about 3 to 4 times that of the influent.

6.3 SOLID WASTE DISPOSAL IMPACT

Promulgation of Federal standards of performance will have little impact on solid waste. If dry control techniques (combustor modifications) are used to meet NO_x regulations, no solid wastes exist. The only potential source of solid waste is from precipitation of solids in the wastes from the water treatment system for the control of NO_x. If these are sewered or returned to the river, no solid wastes result. If they are taken to evaporation ponds, then solid wastes may have to be periodically removed and used as landfill. Most companies contacted during the process of developing emission limitations for stationary gas turbines reported that they either sewered the wastes or returned them to the river. No quantitative data were available for a company using an evaporation pond.

As discussed in section 6.2, the water requirements for a gas turbine using water injection for NO_x control is less than 5 percent of the water requirement for a comparable steam boiler. Therefore, solid waste, if any, would be less than 5 percent of that from the boiler.

6.4 ENERGY IMPACT

The energy impact of Federal standards of performance for stationary gas turbines will be minor. The only energy impact occurs from the use of water or steam injection for control of NO_x emissions. When combustor modifications (dry controls) are used to reduce NO_x emissions, there is no energy impact.

The effects of water and steam injection on power augmentation and gas turbine efficiency were summarized in Table 4.5 and discussed in section 4.5.1.7. A combined cycle gas turbine installation using steam injection suffers about a 1 percent decrease in overall efficiency at a water/fuel injection ratio of 1.0. A simple cycle gas turbine using water injection suffers a 1 percent decrease in efficiency at a 1.0 water/fuel ratio. However, if a small waste heat boiler is added to the simple cycle gas turbine to provide steam for injection purposes, then a net increase in efficiency of about 1.0 percent occurs at a 1.0 water/fuel ratio.

Table 6-25 shows the additional fuel consumption through 1980 if all turbines sold in 1974 through 1980 used water in quantities equal to the

Table 6-25. ADDITIONAL FUEL CONSUMED IN 1980 IF ALL TURBINES IN 1974 THROUGH 1980 USED WATER INJECTION TO MEET REGULATIONS FOR NO_x EMISSIONS

Application	Incremental Sales[1] (MW)		Assumed Average Annual Usage (hours/year)	Heat Rate Btu/KWH	Efficiency Penalty (%) due to water injection[2]	Heat Rate Btu/KWH with water injection	Energy Impact (10^9 Btu)	
	High	Low					High	Low
Utilities	17,670	11,409	2,000[3]	10,300	1	10,729	15,160	9,789
Oil and Gas	2,923	1,051	8,000	13,200	1	13,707	11,860	4,260
Standby	887		80	15,500	1	16,250	53	
Internal power								
a) Oil/Gas	246	37	8,000	13,200	1	13,707	997	150
b) Other Industries	790		8,000	13,200	1	13,707	3,200	
Other Applications	357		2,000	13,200	1	13,707	362	
						Total Btu/year	31,632	17,814
						Equivalent Barrels of oil per day	14,444	8,134

[1] Sales data taken from Table 3-6.
[2] This assumes a 1.0 water-to-fuel ratio.
[3] This is a high estimate since most turbines probably operate less than 2,000 hrs per year.
[4] Assumes 6 x 10^6 Btu/barrel of oil.

fuel flow (1.0 water/fuel ratio) to meet a NO_x regulation. This high water rate and the use of water rather than steam biases the estimated fuel consumption towards the high end since many turbines will use water-to-fuel ratios of much less than 0.5 and, within the next five years, many turbines will incorporate dry control techniques which will obviate the necessity to use water or steam injection. The data indicate that even at a 1.0 water-to-fuel ratio the increase in consumption of fuel oil by the end of 1980 will vary between only 8100 barrels per day and 14,400 barrels per day. The estimated 1980 oil consumption of the United States is 21,000,000 barrels per day and the estimated total energy consumption of the United States in 1980 is equivalent to 47,835,990 barrels of oil per day.[10] When compared to these figures, the additional consumption due to water injection varies between 0.04 to 0.07 percent of the total United States oil consumption and between 0.02 and 0.03 percent of the total U. S. energy consumption.

The data discussed in Chapters 3 and 4 for gas turbines burning low Btu coal gas indicate that emission levels less than 75 ppm at 15 percent oxygen can be met without water or steam injection. Therefore, as users of gas turbines turn to coal gas fuels, the energy impact discussed above will be further decreased.

The other energy impact of using water or steam injection is the impact of the power consumed by the water treatment system. One user of gas turbines reported that this power consumption was negligible when compared to the power consumed by other auxiliaries at gas turbine sites.[11] One source reported that the water treatment system consumes about 20 KW for each 1,000 gallons of treated water.[12] Using this number for the five gas turbines producing 1400 MWH of power as discussed in 4.5.1.2, the power required to operate the water treatment system is 1900 KWH or 0.01 percent of the power generated by the turbines.

6.5 NOISE IMPACT

The use of dry techniques (combustor modifications) or wet techniques (water or steam injection) for the control of air pollution will not impact noise levels. The stationary gas turbine can, unless muffled, constitute a sizeable point source for noise emissions. The application of control technology will not influence these noise levels.

6.6 OTHER ENVIRONMENTAL CONCERNS

6.6.1 Irreversible and Irretrievable Commitment of Resources

Application of wet or dry control systems to stationary gas turbines will result in minimal or no short-term versus long-term trade-offs between environmental parameters. However, the application of wet and dry controls for NO_x results in trade-offs between environmental parameters and energy and the selection of an emission level results in impact on the environment.

As stated in section 6.4, dry controls do not impact turbine efficiency but injection of water at a 1.0 water-to-fuel ratio decreases turbine efficiency by 1 percent which is roughly equivalent to a 3 percent increase in fuel consumption. This one percent decrease in efficiency is higher than that which will be experienced by most turbines since most turbines will require less than a 0.5 water-to-fuel ratio to meet even a 75 ppm NO_x emission level. As shown in Table 6-25, the impact on oil usage if all turbines sold for the years 1974 through 1980 used water injection at a 1.0 water-to-fuel ratio is estimated as 8,100 to 14,400 barrels per day.

Since almost any level of NO_x emissions can be achieved using water injection by simply injecting more water at little additional cost, and available data on dry controls indicates similar trends, the actual emission level for a regulation depends on factors other than the available control technology. Table 6-12 shows the estimated impact of three different NO_x emission levels for all new gas turbines when a seven year period is considered. This table shows that a 75 ppm NO_x emission level will result in decreasing the NO_x emissions over a 7 year period by 650,000 to 980,000 tons while an emission level of 125 ppm will decrease the 7 year NO_x emissions by 330,000 to 440,000 tons. The table shows that at emission levels of 100 ppm and 125 ppm, the NO_x emissions are actually permitted to increase for turbines used in the "oil and gas", "private electric power generation", and "other industrial" categories. This occurs because turbines in these categories now have uncontrolled emission levels lower than the 100 and 125 ppm.

Another alternative which could impact the environment is the implementation of a regulation which allows a delayed standard based on the application and incorporation of dry controls on turbines. If a seven year delay were permitted for all turbines, Table 6-26 shows that the impact on the environment is the emission of 650,000 to 980,000 tons of NO_x, relative to a 75 ppm emission level. If a 75 ppm emission level, based on the use of water injection, were implemented immediately for large gas turbines used in utilities with a seven year delay for smaller turbines to incorporate dry controls, the impact on the environment, as shown in Table 6-26 would be the emission of an additional 38,000 to 68,000 tons of NO_x, relative to an emission level of 75 ppm.

6.6.2. Environmental Impact of Delayed Standards

The environmental impact of delayed standards has been discussed in section 6.6.1 where the impact of a delayed standard based on the application of dry control techniques instead of an immediate standard based on wet or wet plus a percentage of dry reduction was discussed.

TABLE 5-26

ESTIMATED IMPACT OF THREE NO$_x$ EMISSION LEVELS FOR GAS TURBINES WHEN A SEVEN YEAR PERIOD IS CONSIDERED

							CONTROLLED NO$_x$ EMISSIONS And Decrease from Uncontrolled Emission Levels Tons X 10^6					
		Estimated 1974 - 1980 Capacity Increase (millions of HP)[1]	Assumed Operation Hrs/year	Assumed Heat Rate Btu/KWH[2]	Assumed NO$_x$ Uncontrolled Emissions[3] Lb/MWH	Uncontrolled NO$_x$ ~ tons X 10^6	75 ppm @ 15% (0.3 lb/M Btu)	Decrease	100 ppm @ 15% (0.4 Lb/M Btu)	Decrease	125 ppm @ 15% (0.5 Lb/M Btu)	Decrease
Utilities	High	23.7 (17,673 MW)	2000	11,000	12	1.27	0.35	0.92	0.47	0.80	0.59	0.68
	Low	15.3 (11,409 MW)	2000	11,000	12	0.84	0.23	0.61	0.30	0.54	0.38	0.46
	High		500	11,000	12	0.32	0.09	0.23	0.12	0.20	0.15	0.17
	Low		500	11,000	12	0.21	0.06	0.15	0.08	0.13	0.10	0.11
Oil & Gas	High	3.9 (2908 MW)	8000	15,000	5	0.35	0.31	0.04	0.42	-.07[4]	0.52	-.17
	Low	1.41 (1051 MW)	8000	15,000	5	0.12	0.11	0.01	0.14	-.02	0.18	-.06
Private Electric Power Generation	High	1.1 (820 MW)	8000	15,000	5	0.096	0.08	0.016	0.11	-.01	0.14	-.04
	Low	1.1 (820 MW)	8000	15,000	5	0.096	0.08	0.016	0.11	-.01	0.14	-.04
Other Industrial	High	0.5 (373 MW)	8000	15,000	5	0.048	0.036	0.012	0.48	0	0.06	-0.03
	Low	0.5 (373 MW)	8000	15,000	5	0.048	0.036	0.012	0.48	0	0.06	-0.03
TOTALS[5]	High					1.76	0.78	0.98	1.05	0.71	1.30	0.44
	Low					1.10	0.46	0.65	0.60	0.50	0.76	0.33

NOTES

(1) From Table 3-6

(2) (3) From Table 3-8

(4) The negative sign indicates that at this level emissions are estimated to rise over existing uncontrolled levels

(5) Totals include the emissions using 2000 hours/year operation for utility turbines

REFERENCES FOR CHAPTER 6

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7. ECONOMIC IMPACT[1]

7.1 MODEL PLANT SELECTION

7.1.1 Introduction

Section 3.1 discussed the basic structure of the gas turbine industry. As was shown in this section, stationary gas turbines can be and are used in a wide variety of sizes, use a variety of fuels, are located in a wide range of climatic conditions, and are in operations which may produce unique combinations of difficulties in attaining various levels of NO_x control. To characterize better the range of potential impacts, a series of model plants which are typical of existing installations were selected. The selection of these model plants was based on factors considered fairly typical of all turbines in specific applications.

The most important factors in the selection of the model plants were:

(1) capacity or size of the unit;

(2) heat rate characteristics;

(3) location in terms of the proximity of the units to an assured water supply; and

(4) operating hours.

The model plants parallel those applications discussed in Section 3.1. The applications selected for analysis were:

(1) standby power generation;

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[1] All costs presented in this paper are stated in terms of 1975 dollars.

(2) oil and gas compressor stations;

(3) industrial internal power generation;

(4) utility peak load power generation; and

(5) offshore drilling platforms.

Table 7.1 shows the significant characteristics of each of these model plants.

7.1.2 Description of Model Plants

7.1.2.1 Standby Power — Turbines in five size ranges were selected for analysis as standby units. The selection was based on data supplied by Gas Turbine International[1] and are typical of units likely to be used by hospitals, computer facilities, and the telephone industry when a continuous supply of power is necessary. Operating time for these units probably averages around 80 hours/year, with a maximum of 200 hours occurring only in the event of a major power failure. Two hundred hours is considered to be extremely high and not likely to occur very often. It was included in order to bound the potential range of short operating time applications.

In the absence of environmental control, the fuel consumption characteristics (heat rate) of these machines should be about 11,500 Btu's/hp-hr, or 15,500 Btu's/kilowatt hour for a 1,100 hp machine using #2 grade distillate oil.[2]

The large majority of these machines will be located in urban or semi-urban areas where adequate supplies of drinking quality water will be available

7.1.2.2 Pipeline Application — This application is, by far, the most difficult to characterize. As indicated in Section 3.1, relatively slow growth is expected in this area, but the potential for economic impact is great due

TABLE 7.1

MODEL PLANT CHARACTERISTICS

Application	Location	Size	Operating Hours Per Year	Fuel	Heat Rate (Btu/kWh)	Turbine Size (Ft^2)	Capital Cost ($/kW
Standby power	Urban	350 hp	80-200	#2 oil	15,500		
	Urban	1100 hp	80-200	#2 oil	15,500	21	110-1
Industrial							
Internal Power	Plant site	3000 kw 1/	2000/8000	#2 oil	13,200	66	50-7
Pipeline	Compressor station	4000 hp	8000	#2 oil	13,200	66	50-7
Drilling Platform	Offshore	3000 kw 1/	8000	#2 oil	13,200	66	50-7
Utility	Generating station	66000 kw	500 - 8000	#2 oil	10,300	416	80-9

1/ Roughly equivalent to 4000 hp

to potential competition from reciprocating engines. Typical pipeline applications can geographically range from above the Arctic Circle to the Mohave Desert. Onsite water may or may not be available, although one major pipeline company in the Southwest has indicated that no present or planned pipeline is without a water supply.[3]

The machine selected for analysis is a 4,000 hp turbine which may be used in groups of two or three, operating 8,000 hours/year. This particular turbine is of relatively small size, but is commonly used in pipelines. Also, if the imposition of NO_x controls were to have an economic impact, it should show up on the smaller turbines which cannot take advantage of the economies of scale (lower cost per gallon) of the water treatment facilities. Water is assumed to be available onsite or available within fifty miles by transport. The importance of water availability will be discussed in the next section.

Fuel consumption in this application is very dependent on fuel type. Rolls-Royce, which has made more inroads into American markets than any other foreign manufacturer, has recently announced a new turbine with a non-regenerative fuel consumption rate of 7,200 Btu's/hp-hr on natural gas.[4] The turbine made by the major U.S. manufacturer, Solar, has a fuel consumption of 9,750 Btu's/hp-hr, or 13,200 Btu's/kWh, and this rate is not very dependent on fuel used.[5]

If water is available, the turbines selected for pipeline applications are also typical of industrial applications which will act as baseload machine drive or electrical generating applications.

7.1.2.3 Electrical utility applications - At present, the two chief characteristics of turbines used in the electric utility industry are large size and relatively low running time (500-2,000 hr/yr.). Future applications of turbines could, if combined cycles become more frequent, increase in operating time up to medium usage or baseload applications (4,000-6,000 hr/yr.). As discussed in Section 3.1, this application may not become as prominent as once expected due to the high cost of oil and the reduced availabilities of natural gas.

For purposes of this analysis, a 66 MW simple cycle turbine operating 500 to 2,000 hr/yr on #2 distillate fuel was selected. This particular machine is about middle size on the list of machines built by major manufacturers. Also considered was a baseload application (8,000 hr/yr) which might occur under a major breakdown of baseload generating capacity. The turbine selected is designed to produce 66 MW at baseload, but peaks for short time periods at 72 MW.

For purposes of this analysis, a combined cycle system was not considered, because it was felt that the water quality system would be the same for combined and simple cycle machines and the increased capital cost of the combined cycle would only serve to make any impact smaller.

All three utility applications were assumed to be located where adequate supplies of potable water were available.[6]

7.1.2.2.2 Offshore applications — One special application has been broken out for study. Turbines are frequently used on offshore drilling well platforms (up to 21 per platform)[7] for various mechanical drives, gas and oil pumping and electrical power production.

The turbine selected for application was a 4,000 hp machine operating 8,000 hr/yr. Fuel for this application was assumed to be natural gas with a fuel efficiency of 9,000 Btu's/hp-hr.

The primary reason for breaking out this relatively small market for turbines was the potential impact of requiring sea water as feed to water purification equipment.

In summary, six model plants based on common industry applications, considering location. size and fuel mix, were selected. (See Table 7.1) These data will be used in the following sections to determine the economic impact of the proposed standards on the gas turbine industry.

7.2 COST ANALYSIS OF ALTERNATIVE EMISSION CONTROL SYSTEMS

7.2.1 Cost Analysis of NO_x Control for New Turbines

Thermal NO_x, as discussed in Chapter 4, can be controlled in several ways: wet control (water or steam injection), combustor modifications, catalytic cleaning of the tail gas, or some combination of the three techniques. The most prevalent control technique practiced by turbines meeting a stringent state or local requirement is a combination of dry controls plus wet control to bring the emissions down to a 75 ppm level. The emission rate of NO_x without water injection will vary from manufacturer to manufacturer depending on the success and application of dry control techniques. Table 7.2 shows estimated NO_x emission rates for various turbines offered for sale in the United States.

All of the data in Table 7.2 relate to simple cycle machines operating a baseload conditions. Although data are not complete, it appears that regenerative turbines will have higher NO_x emission rates than simple cycle machinies over the entire range of operating conditions.[13]

7.2.1.1 Water Injection - Both the amount and quality of water injected into any turbine are a function of turbine design and will vary from manufacturer to manufacturer and from model to model produced by a company. Table 7.3 lists the quality of water required by the various turbine manufacturers.

TABLE 7.2
EMISSION RATES FOR NON-REGENERATED TURBINES

Manufacturer	Ref.	Size	Emission Rate at Base (ppm @ 15% O_2)
Solar	8	1100 hp 3830 hp	60 - 81 90 - 153
GM	9	3000 hp - 4000 hp	125
Westinghouse	10	66 MW	219
Turbo-Power Machine	10	66 MW	242
General Electric	11	≈60 MW	163
Turbodyne	12	60 MW	340

TABLE 7.3

WATER QUALITY SPECIFICATIONS

	Turbine	Boiler Feed	Electronics
Total Dissolved Solids (TDS) + Non-Dissolved Solids (ppm)	1.0 - 5.0	0.25	0.1
Sodium + Potassium (ppm)	0.5*	0.25	Non-Detectable
Silica (ppm)	0.02	0.0	↓
Particle Size	<10 μ	Not Specified	↓
pH	7.0 - 8.5	6.5 - 7.0	6.6

* Turbo Power and Machine limits sodium to 0.1 ppm.

It is not clear exactly what quality of water is required; the manufacturers' estimates allow for a significant margin of safety. The last column of Table 7.3 shows the quality of water used by a West Coast electronics firm.[14/] Water of this quality is widely used in many applications such as high pressure steam boilers, electronics food and pharmaceuticals. The technology is available to achieve this quality level; however, it will not be required for supplying feedwater for wet control systems. One of the primary costs of NO_x control for turbines is the preparation of water to meet the strict quality specifications. Table 7.4 shows the estimated quantities of water required by the model plants to meet the 75 ppm, 100 ppm and 125 ppm levels. It is important to note that the only model plant which may require additional control to meet the 125 ppm level is the utility application.

The costs estimated in the next sections are based on meeting the 75 ppm level burning liquid fuel. Gaseous fuels typically will result in lower NO_x emission levels and, therefore, will require less control at a lower cost. Section 7.2.1.1.7 discusses the change in control cost if the desired NO_x level were 100 or 125 ppm.

A wide variety of equipment can be used to deliver water of the purity desired. Two main components of a purification system such as this are reverse osmosis and de-ionization. Each process has been described in Chapter 4.

7.2.1.1.1 System selection and cost for model plants — In system selection and sizing, a specific design philosophy was followed. Based on EPA's policy in such areas as flue gas desulfurization and other control equipment where system design and operational capacity are sized for normal operating practice based on engineering judgement, large amounts of redundancy were

TABLE 7.4

ASSUMED FLOW RATES (gpm) FOR ALTERNATIVE STANDARDS

Turbine Size	75 ppm Standard	100 ppm Standard	125 ppm Standard
1100 hp	15	0	0
4000 hp	100	0*	0
66 MW	42	35	25

*Assumes that further limited dry control of the GM machine is possible.

not built into the water purification system. If, for example, a turbine required 60 gallons per minute (gpm) of capacity, a system capable of maintaining 60 gpm would be considered adequate; significant amounts of water storage would not be needed. Alternately, if normal operation of fourteen hours per day required 50,900 gallons per day, a system capable of producing a flow of 35 gpm plus a storage tank of about 20,000 gallons would probably be adeqaute. A similar situation exists in regard to standby units. In any normally occurring operational pattern, these units would not be expected to run more than a few hours at a stretch. Tap water, which can be upgraded using existing water purification equipment, would almost always be available. The circumstances in which these units would run for several days and the water supply would be disrupted, as in the event of a major power failure (perhaps lasting longer than one week), need not be specifically designed for.

Redundancy is often built into the water treatment system to avoid turbine shut down during water treatment system outages. This concern for system reliability does not appear justified. A turbine designed for water injection can operate if the water is not injected into the combustion chamber. Also, the systems required to purify the water for injection are well proven and reliable. San Diego Gas and Electric has reported that in the last one and one-half years of operation, the water system has out-performed the turbine in terms of system reliability.[15]

In this analysis, no effort was made to design an optimal or least cost system. Rather, what was felt to be a workable system with adequate capacity was selected. For example, in some situations a water system designed for continuous application, coupled with a storage tank, could prove to be the most economical. This will vary from site to site and was not considered. Thus, the systems selected are probably less than optimal from a cost standpoint, but adequate to meet the requirements for water quality and water injection rates.

An exception to the above was made in the case of electric utility operation, where actual installed costs for installations using water injection systems were used. It is assumed that, in this case, an effort was made to optimize the overall design.

Special cases of clean water availability or non-availability were also not explicitly considered. An example of clean water availability might be at a utility which has a large amount of built-in redundancy in its boiler feedwater makeup system. If it is possible to use some of this redundancy to supply water to the gas turbine, the incremental turbine control cost would be very low.

A similar series of decision rules are established to select cost data for use in the cost analysis. For each water supply rate required, a series of equipment manufacturers were contacted. Each manufacturer was asked to suggest not only equipment and costs for each flow rate, but also to identify how equipment costs could vary with incoming water supply. These cost estimates were then checked against estimates obtained from users and manufacturers of gas turbines. Whenever possible, any system which appeared consistent with all estimates was used in the model plant analysis. These data were compared with water supply systems actually in use to control NO_x emissions from gas turbines. In all cases, the data used in this analysis are consistent with actual operating experience with water injection NO_x control systems.

7.2.1.1.2 Standby power - The first model plant type selected for study was on emergency power standby units. The two unit sizes selected were a 350 and a 1,100 hp unit. Based on manufacturers' recommendations,[16] 15 gallons per hour was selected for use in the larger unit; in the smaller unit use was scaled down to 6 gallons per hour. However, the possibility exists that these units may not require water injection to meet either standard (See Table 7.2) The low expected operating hours of these turbines have led two manufacturers[17,18] of water purification systems to recommend that portable units be used. The portable system, in one case, consists of a nine-inch diameter tank containing a cation exchange, an anion exchange, and a mixed-bed system. According to the manufacturer,[19] this system will provide water of the required quality at a purchase cost of about $1,200, including controls. The installation cost of the system is less than 15 percent of purchase price. The system should provide about 2000-4000 gallons of water between regenerations in urban areas where United States Public Health Service (USPHS) water quality limits are met. Waters which are higher in dissolved solids would result in shorter intervals between regenerations. Cost per regeneration, about $60-$70, would be incurred probably not more than once per year.* Water from such a system would easily exceed all manufacturers' specifications for quality. Other manufacturers offer similar equipment at both higher and lower costs. One offers a system which, assuming an input water loading of 300 ppm TDS, would provide about 500-600 gallons of capacity (or three months normal operation) at a purchase price of $260 with a regeneration cost of $33.[20] This equipment is laboratory based and may or may not be suitable for industrial applications. The system selected for application to the model plant was the tri-bed system costing

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* If the system provided 3,000 gallons between regenerations, a turbine requiring 15 gallons per hour could be operated 200 hours per year. For this system, the operating cost would be $20/1,000 gallons.

$1,200 plus $200 for installation such as piping, water inlet, etc. An annual operating cost of $60, or $20/1,000 gal, was established based on an estimated one regeneration per year; i.e., if the larger of the two turbines was operated 200 hours per year.

Model Plant	Capital Cost of Water Treatment		Operating Cost		Installation Cost
	Total	$/kW	$/1000 gal	$/yr	
Standby					
350	$1,200	$4.68	$20.00	$9.00	$200.00
1100	1,200	1.50	20.00	22.50	200.00

7.2.1.1.3 Baseload turbine for industrial application - The model plant for this application was a 4,000 hp (3,000 kWh) machine, for which operating time was 8,000 hr/yr.

According to one manufacturer,[21] water injection requirements for this size unit are about 100 gal/hr or 1.67 gal/min. Two different systems were selected for this machine based on two input water qualities. A water very high in TDA, iron and manganese was assumed to be treated by a system consisting of a complete pretreatment operation, including an aeration system, a clarifier and an activated carbon system. This pretreatment was followed by a R.O. unit and two mixed bed de-ionizers, because one unit must be on-line while the other is being regenerated. Alternately, a storage tank could be built and only one DI unit used. Capital costs for this system ranged from $8,000 to $25,000[22, 23, 24, 25, 26] based on manufacturers' estimates.

Most estimates for the system were from \$10-\$12,000. Most manufacturers felt that since this system was smaller than normally built for other applications, unit costs might be high. Data from one turbine user showed an estimated capital cost based on this type of system at \$7,200*.[25]

Installation costs for equipment such as this, from \$400 to \$2,000, are relatively low because the equipment is skid-mounted. Depending on the manufacturer, the necessary equipment would occupy from 10 to 30 square feet.

Operating costs for the above equipment will vary from location to location depending on the requirements for regeneration, the cost of regeneration chemicals, and power for operating the R.O. This cost is estimated by one turbine user as \$1-2/1,000 gal of water produced, or \$864 - \$1,600/year.[28] San Diego Gas and Electric, although operating a much larger system, estimates that the cost will be in this same range of \$1-2/1,000 gal.[29] Manufacturers' estimates were in the same range; one quoted \$4.00/ 1,000 gal or \$3,500/year.[30] Labor costs are not included in the estimates but are expected to be minimal. No more than two hours of labor per week will be required. In unmanned stations, no more than one visit per week might be needed and it coincided with normal site visits.

Sufficient supplies of water may not be available onsite at pipeline or other remote units. Figure 7.1 shows the cost of transporting bulk quantities of water over varying distance,[31] including all costs associated with

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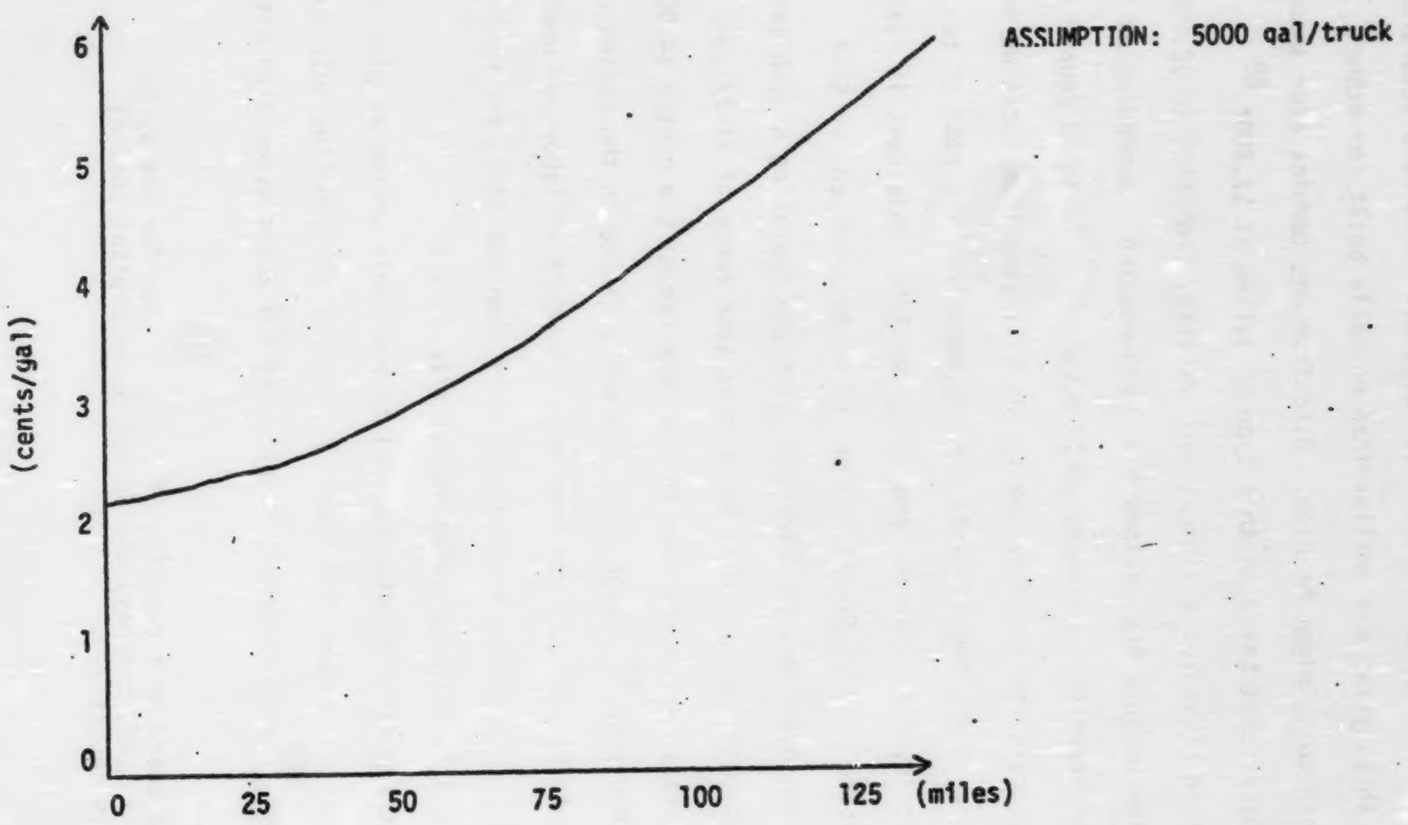
* This number is based on a capacity of 2,400 gal/day for the system (1.67 gal/min x 1,440 min/day) and an estimated capital cost of \$3/gal of daily capacity.

Figure 7.1
WATER TRANSPORT COST



water delivery except unusual detention of the truck at the delivery site. For this analysis, it was assumed that treatment would be done at the site and that a storage tank of 14 day capacity would be provided at an additional capital cost of $18,000.

Model Plant	Capital Cost of Water Treatment		Operating Cost		Installation Cost
	Total	$/kW	$/1000 gal	$/yr	
Pipeline					
4,000 hp	$12,000	$4.00	$3.00	$1,750.00	$2,000.00

7.2.1.1.4 Offshore platform - If the same type of turbine were installed in an offshore drilling well platform, the type of equipment needed to provide a supply of water is quite different from an application where potable water is available. Information provided by one turbine manufacturer[32] estimted the capital and installed cost of distilling sea water would be about $18,000 ($24/1,000 gal). Exhaust heat from the turbine would be used in the distillation process. Automatically vented to the atmosphere, the heat would not represent an issuing penalty. The distilled water would be further treated with a deionizer before being injected into the turbine. Since the quality of distilled water will be higher than normal drinking water, the size of the de-ionizers can be smaller, probably costing in the vicinity of $3,000-$4,000.[33] The costs for operating this type of system are expected to be in the range of $11-12/1,000 gallons.[34] One turbine user,[35] operating a 22 MW regenerative turbine (7 times the size of the model plant and requiring 8 times the water for injection), estimated a capital cost of $82,000 or $4/gal/day of capacity. This cost, which did not include a de-ionizer which may very likely be required, was substantially lower than the $9/a/day ($22,000/2,400 gal/day capacity) capital cost estimate used here. Operating costs for the larger system were also substantially lower, falling in the range of $7.4-9/1,000 gallons. The higher cost estimates for both capital and operating were used throughout the remainder of this analysis.

Installation costs for this equipment have been estimated at $6,000,[36] not including a storage facility. The equipment selected is expected to occupy 80 square feet of deck space.

Model Plant	Capital Cost of Water Treatment		Operating Cost		Installation Cost
	Total	$/kW	$/1000 gal	$/yr	
Offshore					
4,000 hp	$21,000	$7.00	$11.00	$9,640	$6,000.00

7.2.1.1.5 Electric generating station - The model plant selected for analysis in this application is a 66 MW turbine installed at an electric generating station. Operating time is assumed to be 500 to 3,000 hours per year. Water treatment system selection was based on a system currently being installed on a turbine in New Mexico[39] which was designed to meet a 0.3 $lb/10^6$ Btu NO_x limitation, approximately equivalent to 75 ppm on liquid fuel. The system selected by this utility is a R.O. unit preceded by a pretreatment system designed to control the pH. Water used has a TDS of 467 ppm and is not high in other constituents. The R.O. unit is followed by one mixed bed de-ionizer which feeds to a storage tank. Total bid cost for this system delivered to New Mexico was $74,000 adding the $26,000 for the required pumps and storage facilities, system cost is equivalent to $1.50/kW of capacity. The capacity of this system is 60 gpm, somewhat in excess of the 42 gpm requirement estimated by the turbine manufacturer.

Another system on three turbines equalling 192 MW was installed in the spring of 1975 at a total capital cost of $210,000, or 1.09/kW.[38] This system uses a two bed de-ionizer (DI) system in place of a R.O. and mixed unit. A storage tank of 50,000 gallons is provided to serve during regeneration of the DI units. The system is designed to meet an emission rate of 75 ppm NO_x.

San Diego Gas and Electric operates a different type system. Due to the high TDS level of San Diego water, a prefiltering system was required. San Diego also chose to use a dual bed system followed by mixed bed polishers. The system shown, designed for 25,000 gallons per day, or 17 gpm, serves 20 MW of turbines; any excess is used as boiler makeup water. The capital cost of this system was $70,000 in 1971;[39] current operating costs are $2.90/1,000 gallons, excluding amortization of equipment.[40] This operating cost has been assumed for the model plant, although it is felt to be somewhat high. Other costs reported for water systems similar to the one proposed are $1.25/1,000 gal.

Other capital and operating cost have been estimated for expected turbine installations in different utilities. One survey by ASME[41] shows capital costs ranging from less than $1.00 to $10/kW with an average of $4.86/kW. What factors and system types were considered are not clear.

Also unclear are the type of system proposed or the type of input water used. Operating costs for the treatment system ranged from 0.006 mills/kWh to 0.567 mills/kWh, with one at 10 mills/kWh. These operating costs work out to be $.15/1,000 gallons to $14.85/1,000 gallons. Again, the reason for the difference is not clear.

Model Plant	Capital Cost of Water Treatment		Operating Cost		Installation Cost
	Total	$/kW	$/1000 gal	$/yr	
Electric Utility					
66 MW	$100,000	$1.50	$2.90	$7,300	$70,000.00

7.2.1.1.6 Other costs and variables - In addition to the actual costs of the water purification equipment, additional costs are incurred due to the modification of the turbine caused by installation of water injection nozzles and other equipment. These costs, estimated by the manufacturers, vary widely. Table 7.5 shows the costs reported by various manufacturers. The reasons for the difference in reported costs may be due to assumptions that were not specified by the manufacturers. For this analysis, the costs reported by Solar are used for the smaller turbines and the costs reported by Westinghouse in its proposal to New Mexico Electric Company are used for the larger turbine. The choice of Westinghouse over G.E. was made for two reasons: first, the cost for the Westinghouse turbine represents an actual bid cost to modify a turbine for water injection and, second, since it was not clear how much of the equipment included in the water quality treatment system (pumps, piping, etc.) may have been included as turbine modifications by other manufacturers, it was felt best to use the entire cost from one project where possible.

TABLE 7.5
COST OF TURBINE MODIFICATIONS

Manufacturer	Ref.	Size	Cost	$/hp or $/kW
Solar	42	1100 hp	2,900	2.64
		3800 hp	3,900	1.03
G.M.	43	3800 hp	10,000	2.63
G.E.	44	66 MW	70,000	1.06
	45	22 MW	70,000	3.18
	46	22 MW	20,000	.91
Unknown	47	65 MW	53,000	.82
Westinghouse	48	66 MW	45,400	

Turbines located in cold weather climates may require further equipment to prevent freezing of the water. This requirement, however, is not at all certain. All turbines currently in use require heated buildings. If water storage is not required, thermal heating within the building should be sufficient to protect against freezing. If storage is considered a necessity, then the exhaust gas from the turbines could be used; only ducting and venting would add to the cost. Total costs for these precautions are expected to be insignificant if the turbine is used in a baseload application. If the turbine were to be used as an emergency generator or peaking unit to supply electricity, it is assumed that electric heat would be available at minimal additional cost.

The quality of the incoming water could also affect some of the costs reported here. Both capital and operating costs could be significantly impacted if either very poor (both TDS and other constituents high) or better (low TDS) water were available. Estimated changes in capital costs are in the range of ± 15 percent.

A significant variable in assessing capital and operating costs is variation in water injection requirements between manufacturers. Tables 7.6 and 7.7 show that variation.

One other capital cost which could be attributed to an NO_x emission standard, if based on mass of pollutant/unit of power output, is a requirement to measure to power output of each turbine. Based on information supplied by one manufacturer,[49] the installed cost of this equipment on a large turbine is between $13,000 and $17,000. Little, if any, operating and maintenance penalty will be incurred with this equipment. No data have been found concerning the application of this equipment on turbines less than 20,000 hp.

TABLE 7.6
VARIATION IN WATER FLOW REQUIREMENTS FOR 75 ppm EMISSION LEVEL

Manufacturer	Size	Flow (gpm)	Model Plant
Solar	1100 hp	0.25	
	3830 hp	1.66	2
G.M.	3900 hp	0.85	2
Turbodyne	65 MW		
G.E.	66 MW	22	3
	20 MW	5.8	-
Westinghouse	66 MW	42	3
TPM	66 MW	100 gpm	3

* Model Plant:
1 - Standby Power
2 - Baseload application (industrial or pipeline)
3 - Utility power plant

TABLE 7.7

VARIATION IN WATER FLOW REQUIREMENTS FOR 125 ppm EMISSION LEVEL

Manufacturer	Size	Flow	Model Plant*
Solar	1100 hp	0	1
	3830 hp	0	2
G.M.	3900 hp	0	2
Turbodyne	65 MW	50	3
G.E.	66 MW	8	3
	20 MW	4.2	3
Westinghouse	66 MW	25	3

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* Model Plant 1 - Standby
2 - Baseload (industrial or pipeline)
3 - Utility powerplant

7.2.1.1.7 Change in control costs for a 100 or 125 ppm NO_x emission level - Control costs for either the 100 or 125 ppm emission level are expected to be quite similar. No additional control will probably be required for the standby or industrial turbines. The flow rate reduction required for the utility application is not expected to significantly affect the overall control cost for those manufacturers who would require wet type controls to achieve either the 100 or 125 ppm emission limit.

7.2.1.1.8 Summary of water injection control costs - Table 7.8 shows a summary of the cost data selected for the model plants.

7.2.1.2 Steam Injection - As also discussed in Chapter 4, the injection of steam into the combustion chamber can successfully control emissions of NO_x. Injection of steam (on a mass comparison with water) may not be as efficient as water injection in reducing NO_x emissions; up to three times as much steam might be required, according to one turbine manufacturer.[50] On the other hand, other manufacturers report equal effectiveness for water and steam. Steam injection results in about a 2 percent improvement over water injection in the heat rate of the turbine and a comparable increase in power output[51] The drawback to steam injection is that, unless a ready supply of steam is available, the cost for raising steam to the proper temperature and pressure is considerably higher than the costs for providing water for injection. According to one turbine manufacturer,[52] this cost could be three times higher than the cost of water injection, assuming the installation of an intermediate pressure boiler specifically for the injection steam.

In certain circumstances the injection of steam seems to have potential applications. If the turbine is part of a combined cycle operation, it is possible to take some of the steam-from the boiler portion of the combined cycle plant for use as the steam input. Of course, this would detract from the overall efficiency of the combined cycle, but in many instances the overall

TABLE 7.8

SUMMARY OF MODEL PLANT COSTS FOR WATER INJECTION FOR A 75 ppm EMISSION LIMIT

Turbine Size	Application	Capital Cost of Water Treatment Facility		Operating Cost of Water Treatment System		Turbine Modification Cost	Installation Cost	Space (f
		Total	$/Kw	$/1000 gal	$/Yr			
350 hp	Standby	1200	4.68	20.00	9.00	1000	200	--
1100 hp	Standby	1200	1.50	20.00	22.50	2900	200	--
4000 hp	Pipeline[1/]	12000	4.00	3.00	1752.00	3900	2,000	20
4000 hp	Offshore	21000	7.00	11.00	9636.00	3900	6,000	80
66 MW	Utility -	100000	1.50	2.90	7300.00	45,400	70,000	--

1/ For water transport cost see Figure 7.2; add $18,000 cost for a 14-day storage tank.

economics may justify the use of the steam. Steam produced by other onsite boilers either at an electric utility or at other installations may also be used for steam injection. For example, one turbine user has recommended the use of steam injection solely to enhance output.

In summary, steam injection is an effective alternative to water injection which may be used in circumstances where the particular site economics would justify the choice. None of the model plants have been assumed to use steam injection, since it is not felt that steam injection would be used unless particular site economics justified the choice.

7.2.1.3 Catalytic Exhaust Gas Treatment Control - One vendor has developed a catalytic exhaust gas cleanup control device to remove NO_x from turbine exhausts.[53] As in the case of steam injection, this device appears to have use in certain applications, especially where a fuel high in fuel-bound nitrogen was used or where water was not available.

Estimated costs for this system applied to a 30 MW turbine are in the vicinity of $135,000 ($4.50/kw) which is higher than water injection costs for a similar sized machine (i.e., approximately $1.5-$2.0/kw). In addition, the system requires the use of considerable amoun-s of ammonia and a method of storing and handling it. Also, operation of the system causes about a 2 percent decrease in turbine output.[54]

The catalytic exhaust cleanup control system is probably suited for certain applications. Its costs and operating characteristics are not presently more favorable than the costs of a water injection system; for purposes of economic analysis, it will not be considered as a separate option, since it is not expected to be used unless the cost is less than the cost of a water injection system.

7.2.1.4 Dry Controls - By modifying the design of the combustor or by revising the methods of firing, lesser amounts of NO_x can be formed. This method of control is called "dry" control technology, since it does not require the addition of water or steam to meet NO_x limitations. To varying degrees most manufacturers have incorporated dry controls and have reduced gas turbine emissions relative to uncontrolled levels. Some machines (see Table 7.2) have reached the level of the most stringent standard and machines from at least one manufacturer in each size range can meet the 125 ppm level using the current level of dry control. Some of the smaller machines on the other hand, do not incorporate dry controls, but they may meet emission levels due to low firing temperatures and compression ratios.

Many gas turbine manufacturers felt that if current R&D programs were successful, future improved dry control sufficient to meet the most stringent proposed standard should cost about the same as current levels of water injection. Therefore, it was assumed that no additional dry control was needed to comply with the proposed standard levels. Rather, as in the case of steam injection, water injection was assumed for the model plants because it was felt that any added degree of dry control must cost less than current water injection.

7.2.1.5 Effect of Fuel Specifications in NO_x Control -- The predominant fuel currently used by gas turbines is either distillate fuel oil or natural gas. In fact, between January and April 1975, no utility turbine in the U. S. burned any fuel except distillate or natural gas. As discussed in Chapter 4, these trends will probably continue in the future.

Both natural gas and distillate fuel oil are relatively clean fuels which are low in both fuel nitrogen and sulfur. Sulfur will be discussed in the next section. Fuel bound N_2 is capable of significantly impacting the ability and cost of turbine compliance with NO_x limitations. For example, a fuel containing 0.1 percent N_2 (very high for distillate and average for residual) would add about 32 ppm of NO_x to a turbine burning 100 gpm of fuel; a fuel containing 0.046 percent N_2 (higher than almost all distillates) would add about 16-17 ppm to the stack gases.[55] Wet controls can compensate for these levels, although the water or steam injection rate would have to increase. There should be no problems with catalytic exhaust cleanup control. If relatively clean fuels are used, fuel bound N_2 should not be a problem.

If turbines were to burn residual fuel oil, a cleaning facility to remove sodium, ash and other water soluble impurities would have to be installed to treat the oil prior to its use in the turbine. Based on discussions with two manufacturers of this equipment, the cost to treat residual fuel for a turbine using 100 gpm (about 60-70 MW) was between $400,000-750,000 plus installation, estimated at a minimum of 50 percent of the capital cost. Operating costs would not be expected to be high ($2/1,000 gallons) except where large amounts of vanadium were present. One manufacturer quoted a price of $20-25,000/day for a very large turbine installation burning a high vanadium residual oil. These costs do not include any additional costs which might be required to provide facilities for handling residual fuel oil. These facilities might include heavier oil pumping systems, turbine modifications, oil heaters, etc. which may already be present if the turbine is located at an oil-fired power plant. These costs were confirmed by several gas turbine users who reported installed costs for fuel treatment systems ranging from $1,300,000 to $1,700,000.[56,57] Based on the above cost data both manufacturers of oil treatment facilities indicated that they did not foresee a large amount of residual fuel being used to fire turbines in the United States. To confirm this assessment, a simplified discounted cash flow analysis was performed using the following equation:

$$PV = \sum_{t=1}^{n} \frac{(1-T)(R_t - C_t - D_t) + D_t - K_t}{(1 + r)^t}$$

Definition of Terms

K = capital cost and depreciable installation costs

R = gross revenues

T = applicable tax rate

C = operating costs, overhead charges, working capital requirements

D = average or annual rate of depreciation

t = year of cash inflow or outflow

n = useful life of investment

r = discount rate (i = internal rate of return when PV = 0).

By setting the present value equal to zero, this equation can be solved for the price differential between residual and distillate fuel oil required to justify the investment. The formula was solved using the following assumption and for two levels of operation (i.e., 500 and 2000 hours per year):

K = \$1,000,000

C = \$2/1000 gallons: \$4704 @ 500 hours and \$18816 @ 2000 hours

D = straightline over 20 years = \$50,000

T = .5 (federal 48%, state 4%)

r = 15% (present worth factor = 5.8715)

R = solve for

At 500 hours of operation per year, the required price differential would be roughly $5.22/barrel or $.126/gallon. At 2000 hours per year the differential decreases to $1.38/barrel or $.033/gallon. Historical price differentials between residual oil and distillate have been about 2-3¢/gallon at the refinery[58] and currently are about 2¢/gallon between high sulfur residual oil and distillate oil.[59] Given the additional costs of using residual fuel oil not considered in the above analysis, such as the cost of revised storage facilities, revised unloading and fuel handling facilities, plus the effects of residual oil on the turbine itself, it appears that the outlook for residual fuel oil use in turbines is very limited, at least in the near term.

The use of synthetic fuels in gas turbines is being actively researched at this time. Of special interest is the use of a low Btu coal gasification system coupled with a combined cycle turbine. Preliminary data indicates that a low Btu gas can be burned in a turbine with low NO_x emissions and possibly at a cost equal to or less than current operations.

At the present time, the use of this fuel cannot be considered a NO_x control technique but rather an attempt to increase supplies of acceptable fuels for turbine use. The use of low Btu gas appears to be able to meet either NO_x emission level (75 or 125 ppm) at no additional cost and may offer opportunities to increase turbine efficiency and temperatures while remaining within NO_x limits.

Other fuels such as methanol have also been mentioned as possible gas turbine fuels. As with low Btu gas from coal, much of the interest in these fuels comes not from a NO_x control viewpoint, but from an effort to assure adequate or reasonably priced fuels for turbines. Most of the other fuels will meet either NO_x emission level with no water injection or additional costs.

7.2.2 Cost of Control of Sulfur Oxides for New Turbines

The only source of sulfur oxides from a gas turbine is sulfur contained in the fuel. As discussed in Section 7.2.1, few if any turbines the U. S. are expected to burn residual. In this section, the costs attributable to burning 0.28% sulfur oil (about 0.3 lbs/10^6 Btu) versus 0.74% sulfur oil (about 0.8 lbs/10^6 Btu) are analyzed. As will be discussed, the 0.28% sulfur level is typical of most distillate fuel oils, while the 0.74% level represents the New Source Performance Standard for oil-fired steam generators.

7.2.2.1 Residual Fuel Oil -- Costs for residual oil desulfurization are dependent on the type of crude used to produce the residual fuel oil. Tables 7.9 and 7.10 show the costs for various crude types and desired sulfur levels. Based on a previous study of the electric utility industry,[60] a distribution of imported product by source and demand by sulfur content were developed. These are shown in tables 7.11 and 7.12. The demands shown are based on current State regulations and Federal new source performance standards. The average cost to reach these levels was $0.99/bbl or 2.4¢/gal of fuel; the average required sulfur content was 0.72%.

The actual cost of either sulfur in fuel limitation on any particular gas turbine is very difficult to determine. The cost over State requirements will vary from location to location. On the average no incremental cost over and above State requirements is expected if sulfur in residual oil were limited to 0.74%. The potential range of the costs might be up to $1-1.40/bbl for the 0.74% requirement and $1.50-$2.00/bbl for the 0.27% limit, if the State requirements were lenient.

For the model electric utility plant of 66 MW operating 1,000 hours per year, this amounts to an increased fuel cost of $230,000 on a 13 percent increase in fuel costs as a maximum to meet a 0.27% sulfur requirement. This is based on a $2.00/bbl desulfurization charge. Actual expected costs over

TABLE 7.9

DIRECT DESULFURIZATION COSTS
(1974 DOLLARS PER BARREL)*

Desired Sulfur Level	Crude Source	
	Arabian Light	Texas Sweet
0.2%	1.73-2.00	1.23
0.3%	1.60	1.13
0.5%	1.46	0.96
0.7%	1.33-1.77	0.74
1.0%	1.18	0.49
1.5%	0.90	0.00
2.0%	0.61	0.00
2.5%	0.35	0.00
3.0%	0.00	0.00

* Reference 60

TABLE 7.10

INDIRECT DESULFURIZATION COSTS*
(1974 DOLLARS PER BARREL)

Desired Sulfur Content	Oil Source	Venezuela			California		
	Process	Visbreak** + Vacuum	Coking HDS	Heavy Oil Crack	Visbreak** + Vacuum	Coking HDS	Heavy Oil Crack
0.2%		$1.36	$1.64	$1.66	$1.48	$1.73	$1.79
0.3%		1.28	1.54	1.63	1.34	1.64	1.77
0.5%		1.15	1.36	1.50	1.33	1.54	1.73
0.7%		0.95	1.18	1.36	1.15	1.41	1.64
1.0%		0.81	0.92	1.14	1.08	1.24	1.43
1.5%		0.47	0.52	0.59	0.79	0.96	1.00
2.0%		0.13	0.15	0.17	0.55	0.67	0.69
2.5%		0.00	0.00	0.00	0.31	0.37	0.38
3.0%		0.00	0.00	0.00	0.00	0.00	0.00

* Reference 60.

** Costs reflect discard of high-sulfur pitch.

TABLE 7.11

MIX OF CRUDE OILS USED TO PRODUCE RESIDUAL OILS IN U.S. MARKETS (MILLIONS OF BARRELS PER DAY)*

From Imported Product	2.13	
Venezuela		(1.92)**
Arabia (light)		(.21)
From Domestically Produced Crude	.87	
Texas (sweet)		(.78)
California (sour)		(.09)
From Imported Crude	.70	
Canadian/Algerian***		(.18)
Arabia (light)	—	(.52)
Total Demand	3.70	

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* Reference 60.

** Numbers in parenthesis refer to the ambient of oil (millions of barrels per day)

*** Approximately equivalent to Texas sweet.

TABLE 7.12

DISTRIBUTION OF RESIDUAL FUEL OIL* DEMAND IN 1980

Sulfur Content %	Demand (10^6 BBL)	Percentage	Cumulative Percentage
0 - 0.2	1.64	0.2	0.2
0.2 - 0.4	97.43	13.2	13.4
0.4 - 0.6	217.9	29.6	43
0.6 - 0.8	114.5	15.5	58.5
0.8 - 1.2	196.5	26.6	85.1
1.2 - 1.8	3.2	0.4	85.5
1.8 - 2.2	28.9	3.9	89.4
2.2 - 2.8	13.3	1.8	91.2
>2.8	63.8	8.8	100

* Reference 60.

State requirements are expected to be much smaller than this. Current costs for low sulfur (0.3 %) oil are about \$1.30/bbl over high sulfur oil.[61/]

7.2.2.2 Distillate Fuel Oil -- The majority of distillate fuel oils sold in the U.S. will meet both a 0.3 and 0.74% sulfur regulation with no further treatment. Some fraction of the distillate, however, may have to be desulfurized prior to use if a 0.3 standard were imposed. Table 7.13 shows the distribution by sulfur content of distillate fuel delivered to power plants in 1974.[62/]

Based on the above data, it is clear that the economic impact of a 0.74% limitation is effectively zero, while some desulfurization could be required if a 0.3 standard were set.

As the potential demand for low-sulfur distillate oil by turbines is a relatively small proportion of total demand for middle distillates, and since a large proportion of this oil is below 0.3% sulfur, the demand can probably be met by blending distillates or by selectively purchasing low-sulfur crude. The maximum premium for this fuel can be estimted as the cost to desulfurize a higher sulfur distillate down to a 0.3 level. Little data exist on desulfurizing distillate fuel, but the cost can be approximated closely by the cost to desulfurize gasoline.

Desulfurization of gasoline can be used as a proxy for desulfurization of distillate because the higher (> 0.3%S) sulfur distillate is produced as a stream from the catalytic cracker and gasoline is desulfurized by treating the feed to the cat cracker. The estimated cost for this is about 1¢/gal.[63]

If, as has been suggested, gasoline is desulfurized to alleviate sulfuric acid from automobiles, the cost attributable from low-sulfur distillate would be reduced to zero.

Overall, the maximum differential impact between a 0.3 and 0.75% sulfur limitation on distillate fuel oil is expected to be less than 0.5¢ gal, or less than a 1%*increase in fuel costs for turbines.

TABLE 7.13

DISTRIBUTION OF DISTILLATE OIL BY SULFUR CONTENT - 1974*

Sulfur Content (%)	Percentage of Total	Cumulative
< 0.1	1.9	1.9
0.1 - 0.2	45.4	47.3
0.2 - 0.3	12.6	59.9
0.3 - 0.5	25.4	85.3
> 0.5	14.2	100.00

* Reference 60.

7.2.2.3 Flue Gas Desulfurization (FGD)

-- In theory, it should be possible to apply FGD systems to turbine in a fashion similar to power plants. In practice, the gas flow rate of a turbine relative to a steam generating power plant is extremely high. The estimated cost of a FGD system can be estimated on the basis of the gas flow rate to the system. The flow rate of the model electric utility turbine is about 1.5 x 10^6 ACFM or equivalent to a 500 MW steam generator. Installed costs for this size are estimated at between 26 and 30 million dollars or 2 to 3 times the cost of the turbine.[64/] Needless to say, it is not expected that many turbine users would decide to use FGD systems.

7.2.3 Cost of Particulate Control for New Turbines

As discussed in Chapter 4, a wet scrubbing system has been developed to reduce particulate emissions for gas turbines. Based on data from the manufacturer[65] the installed cost of this system has been estimated for the model plants. Table 7.14 shows these results. It is obvious that for the smaller turbines the cost of this equipment is prohibitive. For the utility application the capital cost is quite high but other factors must also be considered. Makeup water use by the system is about 500 gpm; the space requirement is very large for the cooling tower alone and, coupled with the space needed for the scrubber itself, can amount to over twice the space needed for the turbine itself.

7.2.4 Cost of Control for Modified Turbines

As discussed in Chapter 5, the probability is quite high that few, if any, turbines will fall within the definition of modifications. However, if any turbines did fall within the modification definition, the cost of control would not be different than for new turbines, since the area for control (either wet or dry control) is within the combustor can which is commonly changed during any type of change which might be considered a modification.

TABLE 7.14

COST OF PARTICULATE CONTROL FOR MODEL PLANTS

Model Plant	Application	Installed* Cost of Scrubber	Area Required** (ft^2)	Power Required	Base Turbine Installed Cost	% Increase in Turbine Cost
1100 hp	Standby	$220,000	150	20 KW	$ 180,000	122
4000 hp	Pipeline	$290,000	300	60 KW	$ 312,000	93
66 MW	Utility	$970,000	2500	500 KW	$8,000,000	15

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* Includes scrubber, cooling tower, clarifier and engineering expenses.

** Cooling tower only; additional 12 ft. around stack required for scrubber.

7.3 COST ANALYSIS OF GAS TURBINE EMISSION CONTROL SYSTEMS

7.3.1 Introduction

The cost estimates presented below cover four major gas turbine applications:

(1) standby emergency power generation units;

(2) industrial uses such as internal power generation or pipeline compressor stations;

(3) off-shore drilling platform pumping and compressor units; and

(4) utility peak load generating stations.

The purpose of this section is to analyze the potential impact of the proposed standards of performance on the cost of generating power in these applications. To the extent possible, the data used in estimating incremental control costs were taken either from data provided by users of gas turbines currently using water injection to control NO_x emissions or from estimates provided by manufacturers of gas turbines and manufacturers of water purification equipment. These data were subjected to detailed analyses and are considered to be reliable. However, since these data were estimates, the parameter values were widely varied to determine the sensitivity of the resulting aggregate cost estimates to changes in the value of specific subcomponents of the emission control system.

A mathematical model was constructed to assist in estimating the cost (mills/kilowatt hours) and percentage impacts of the proposed standards of performance. The model was also used to test the sensitivity of the estimates to changes in some of the underlying assumptions. Contained within the model were a number of parameters which impacted directly on the unit costs (mills/kilowatt hours) of operating gas turbines. Some of these parameters may be considered to be independent of the application; others depend on the location, size and use to which the turbine will be put.

7.3.2 Baseline Cost Estimates

Gas turbines are widely used as standby generating stations by hospitals, telephone companies and other institutions where power interruptions cannot be tolerated, by industry for internal power generation, by pipelines at compressor stations and by the utility industry to meet peak load requirements. In all cases, the cost of operating the turbine is a function of the application. Baseline costs for each of the model applications are provided below. The costs have been estimated in terms of fixed capital charges, fuel costs and operating and maintenance expenditures.

Although some states and localities have regulations requiring NO_x control on gas turbines, baseline costs were estimated assuming no existing control requirements, because few specific regions have control requirements for gas turbines. In addition, since the assumption tends to overstate the incremental cost, it can be viewed as conservative.

7.3.2.1 Parameter Values

7.3.2.1.1 Fixed capital charges -- The fixed capital charge is the percentage of the total capital cost including all depreciable installation charges which must be recovered each year over the life of the investment to provide an acceptable return on invested capital. The actual percentage which must be recovered will depend on the type of cost accounting system used, the method used to depreciate capital stock for tax purposes and whether installation and other relatively fixed costs are included with the purchase price of the equipment. Typical fixed capital charges used by industry vary from 15 to 25 percent for plant and equipment expenditures with useful lives in excess of ten years.

In the cost model, three fixed capital charge rates were used. Twenty percent was used as an average value in most instances. To test the sensitivity of the result, the charge rate was varied from 15 to 25 percent. It is doubtful that a fixed charge rate higher or lower than these values would be appropriate given the cost of funds in the existing capital market and the useful life of the equipment.

7.3.2.1.2 Fuel costs -- Depending on the application, fuel costs may represent the most important element in the overall cost structure for gas turbines. Clearly, this will not be the case for emergency standby units which are seldom, if ever, operated more than 200 hours per year. Fuel costs in this application probably never account for more than 20 percent of the annual total cost of the system. In almost all other applications, with the possible exception of utility peaking plants operated less than 1,000 hours per year, fuel costs will account for over 50 percent of annual costs.

Two fuel costs were used in this analysis. Distillate fuel oil, which accounted for roughly 60 percent of the energy consumed by the electric utility industry in turbines in 1974, was priced at $2.18 per million BTU. This price, based on the current selling price of distillate fuel oil in the Gulf Coast region of the country,[66] is roughly equivalent to $12.70 per barrel, or $.30 per gallon. The other fuel cost was the average city-gate price for natural gas which, during the first quarter of 1975, was equivalent to $.66/MMBTU.[67/]

With regard to the cost evaluation, in most cases the distillate fuel oil price was applied for two reasons. First, natural gas is in short supply. Industrial customers, on firm as well as interruptible contracts, are being severely curtailed. With the exception of electric utilities located in the intrastate market, the outlook for future natural gas supplies in the utility industries is even less favorable. It is unlikely, therefore, that new turbine units will be brought online using natural gas as their primary fuel. More likely, new turbines that have gas-firing capabilities will be equipped to fire oil in the event of future curtailment. Secondly, in intrastate markets where natural gas is still available, the price is approaching or even exceeding the price of oil. Table 7.15 shows a summary of intrastate natural gas contracts signed during the period January 1, 1974 to January 9, 1975. While in the quoted new contract prices vary considerably, they are considerably higher than the controlled interstate gas prices and likely to continue to increase. In a free market, natural gas prices can be expected to increase to levels comparable to substitute fuels. The only major fuel which is clearly substitutable for natural gas in gas turbines is distillate fuel oil.

7.3.2.1.3 Operating and maintenance charges -- Gas turbines have relatively low operating and maintenance requirements compared to other equipment. This is one of their primary advantages. Data from the Federal Power Commission indicate that for utility systems operating and maintenance costs, exclusive of fuel costs, are generally less than one percent of the installed capital cost of the turbine.[68] Data on a randomly selected group of utility turbine installations are presented in Table 7.16. In seven of the nine

TABLE 7.15

NATURAL GAS INTRASTATE

GAS PRICES

(Jan. 1, 1974 - Jan. 9, 1975)

FPC Area	PRICE ¢/MCF High	Average	Volume Mcf
Appalachian-Illinois 1/	75.00	52.95	7,164,886
Other Southwest 2/	152.53	116.93	24,251,836
Southern Louisiana 3/	121.19	87.99	25,031,297
Texas Gulf Coast 4/	207.22	83.09	256,864,228
Permian Basin 5/	146.79	89.63	185,258,708
Hugoton-Anadarko 6/	110.60	73.99	56,770,320
Rocky Mountain 7/	77.00	27.43	3,108,302
Other Areas 8/	243.89	131.76	3,941,964

Source: Statement of John Nassikas, FPC, before the House Committee on Interstate and Foreign Commerce, July 14, 1975.

1/ Areas of New York, Pennsylvania, Ohio, West Virginia, Maryland, Virginia, Kentucky, Illinois, and Indiana.

2/ Areas of Mississippi, Louisiana, and Texas.

3/ Southern Louisiana.

4/ Areas of Texas.

5/ Areas of Texas and New Mexico.

6/ Areas of Texas, Oklahoma, and Kansas.

7/ Areas of Louisiana, Utah, New Mexico, Colorado, Wyoming, Nebraska, Montana, North Dakota, and South Dakota.

8/ The remainder of the Continental United States not included in any of the other producing areas.

TABLE 7.16

UTILITY OPERATION AND MAINTENANCE EXPERIENCE WITH GAS TURBINES

Plant Name	Unit Size (MW)	Installed Cost ($/KW)	Operating and Maintenance Expenditure	
			% of Installed Cost	Mills/ KwH
Simple Cycle				
Yuma	192.0	$ 85	1.4	3.3
Intercission City	340.0	$ 73	0.5	0.7
Turner	181.0	$ 86	0.5	1.3
Fox Lake	29.6	$ 68	0.3	2.2
Sycamore	157.5	$ 89	0.2	1.0
Meramec	58.9	$ 91	0.3	2.8
Combined Cycle				
Sterlington	523.4	$102	2.7	.8
Croydon	546.0	$115	.2	.3
Gilbert	205.2	$160	.7	.4

Source: Form 12, FPC data for 1974.

cases, operating and maintenance costs were less than 1 percent of the installed capital costs when utility systems were considered.

For industrial applications, a higher average operating and maintenance cost was used, because these units are generally operated continuously and require a major overhaul every five or six years.[69/] The cost of the overhaul ranges from 15 to 25 percent of the installed cost of the turbine. For purposes of cost evaluation, a 5 percent operating and maintenance cost per year was assumed. Four percent is attributable to the required overhaul and 1 percent for general operating and maintenance expenses.

Units in standby service generally receive little maintenance. In the absence of a power failure, they may be activated once or twice a year to ensure that they will operate properly in the event they are needed. Since data on actual operating experience were lacking, operating and maintenance costs were assumed to be approximately 1 percent of the installed purchase price.

7.3.2.2 Cost Estimates -- The parameters discussed above were used to establish baseline costs for the model plant applications. Table 7.17 shows the estimated costs on a mills per kilowatt hour basis. Heat rates (BTU/kwh) and installed costs per kilowatt were based on manufacturers' specifications and data reported to the FPC.[70/]

The cost estimates presented in Table 7.17 vary within wide margins. Standby applications have the greatest variability, with costs ranging from 265 to 1189 mills per kilowatt hour. Capital costs dominate, accounting for over 80 percent of the cost per unit of output in all cases. Fuel costs are, for the most part, quite small compared to other gas turbine applications.

TABLE 7.17

BASELINE COST ESTIMATES[1]
FOR MODEL PLANT APPLICATIONS

A. Standby Units

1. Operating Parameters

Unit Size - 350 hp, 1100 hp
Installed Capital Cost - \$220/Kw
Heat Rate - 15,500 BTU/Kwh
Fixed Capital Charge - 20% of Installed Cost
Operating and Maintenance - 1% of Installed Cost
Fuel Costs - \$2.18/MMBTU (#2 oil)

2. Estimated Baseline Costs

	Operating Hours -- per Year				
	40	80	120	160	200
Mills/Kwh	1188.8	611.3	418.8	322.5	264.8
Allocation of Cost (% Total)					
(1) Fixed Capital Charges	92.4	90.0	87.6	85.2	83.0
(2) O + M	4.6	4.5	4.4	4.3	4.2
(3) Fuel Cost	2.8	5.5	8.0	10.5	12.8
Total	100.0	100.0	100.0	100.0	100.0

B. Industrial Applications

1. Operating Parameters

Unit Size - 3000 Kw
Installed Capital Cost - \$120/Kw
Heat Rate - 13,200 BTU/Kwh
Fixed Capital Charge - 20% of Installed Cost
Operating and Maintenance - 5% of Installed Cost
Fuel Costs - \$2.18/MMBTU (#2 oil)

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[1] All costs are estimated on a before tax basis.

TABLE 7.17 CONTINUED

2. Estimated Baseline Costs

	Hours of Operation per Year				
	2000	3000	4000	6000	8000
Baseline Costs (Mills/Kwh)	43.8	38.8	36.3	33.8	32.5
Allocation of Cost (% Total)					
(1) Fixed Capital Charges	27.4	20.6	16.5	11.8	9.2
(2) O & M	6.9	5.2	4.2	3.0	2.3
(3) Fuel Costs	65.7	74.2	79.3	85.2	88.5
Total	100.0%	100.0%	100.0%	100.0%	100.0%

C. Utility Application

1. Operating Parameters

Unit Size - 66,000 Kw
Installed Capital Cost - $150/Kw
Heat Rate - 10,300 BTU/Kwh
Fixed Capital Charge - 20% of Installed Cost
Operating and Maintenance - 1% of Installed Cost
Fuel Costs - $2.18/MMBTU (#2 oil)

2. Estimated Baseline Cost

	Hours of Operation per Year				
	200	500	1000	2000	8000
Baseline Costs (Mills/Kwh)	180.0	85.5	54.0	38.2	26.4
Allocation of Cost (% Total)					
(1) Fixed Capital Charges	83.3	70.2	55.6	39.3	14.2
(2) O & M	4.2	3.5	2.8	2.0	.7
(3) Fuel Cost	12.5	26.3	41.6	58.7	85.1
Total	100.0%	100.0%	100.0%	100.0%	100.0%

Baseline costs for the industrial applications were the least sensitive to changes in operating hours, since fuel costs dominated and, to a significant degree, the heat rate was independent of the number of hours of operation. Baseline costs ranged from 43.8 mills per kilowatt hour for a turbine used in a single shift industrial operation (2,000 hrs/yr) to 32.5 mills per kilowatt hour for a unit operated continuously eleven months (8,000 hrs) of the year.

Simple cycle turbines are used by utilities and generally confined to peaking operations. Few of these units are operated more than 2,000 hours per year. For a unit operated 500 hours per year, the baseline cost would be approximately 86 mills per kilowatt hour. In the extreme case of baseloading a simple cycle gas turbine, the cost would be approximately 26.4 mills per kilowatt. Capital charges and fuel costs in the 500 to 2,000 hour operating range are equally important.

7.3.2.3 Sensitivity Analysis - The baseline cost estimates are quite sensitive to changes in the operating parameters. In order to test the different sensitivity, estimates were prepared assuming values for the underlying parameters. These estimates were used to provide bounds around the baseline estimates developed above. Table 7.18 shows the resulting cost estimates and the assumed values for the operating parameters.

Although standby units are generally more sensitive to changes in capital costs and fixed capital charges, at higher operating hours fuel costs become more important. The industrial examples are quite sensitive to changes in fuel prices. At the controlled interstate gas price of $.66/MMBTU, costs

TABLE 7.18

BASELINE COST SENSITIVITY ANALYSIS

Standby Unit

	80 hr/year		200 hr/year	
	Mills/Kwh	%	Mills/Kwh	%
Baseline Estimate	611.3	--	264.8	--
Fuel				
. $.66/MMBTU	587.7	-3.9	241.2	-8.9
High Capital Cost				
. 270/Kw	742.5	21.5	317.3	19.8
Fixed Capital Charge				
.25%	748.8	22.5	319.8	20.8
.15%	473.8	-22.5	209.8	-20.8
Range	473.8 to 748.8	-22.5 to 22.5	209.8 to 319.8	-20.8 to 20.8

Industrial Application

	2000 hr/year		8000 hr/year	
	Mills/Kwh	%	Mills/Kwh	%
Baseline Estimate	43.8		32.5	
Fuel				
. $.66/MMBTU	23.7	-45.9	12.5	-61.5
High Capital Cost				
. 150/Kw	47.5	8.4	33.4	2.8
Fixed Capital Charge				
. 25%	46.8	6.8	33.3	2.5
. 15%	40.8	-6.8	31.8	-2.5
Range	23.7 to 47.5	-45.9 to 8.4	2.5 to 33.4	-61.5 to 2.8

TABLE 7.18
CONTINUED

Utility Application

	500 hr/year Mills/Kwh	500 hr/year %	2000 hr/year Mills/Kwh	2000 hr/year %
Baseline Estimate	85.5		38.2	
Fuel				
. $.66/MMBTU	69.8	-18.4	22.5	-41.1
High Capital Cost . $180/Kw	98.1	14.7	41.4	8.4
Fixed Capital Charge				
. 25%	100.5	17.5	42.0	9.9
. 15%	70.5	-17.5	34.5	-9.9
Range	69.8 to 100.5	-18.4 to 14.7	22.5 to 42.0	-41.1 to 9.9

per kilowatt hour decrease by almost two-thirds. The industrial unit was much less sensitive to changes in capital costs. For example, a 25 percent increase in the costs per kilowatt only changed the cost per kilowatt hour by 2.4 percent for the single shift operation and by 2.8 percent for the baseload application. The results on the utility side are mixed. If the unit is operated solely as a peaking plant, all five changes appear to have the same relative impact, varying between plus or minus 20 percent. If the unit is operated more as a shoulder peaking plant (intermediate load at approximately 2,000 hours per year), changes in fuel costs dominate.

In summary, depending on the assumptions regarding operating parameters, the costs per kilowatt hour may range anywhere from 611.3 mills per kilowatt hour for the standby unit operating 80 hours per year, to 12.5 mills per kilowatt hour for the baseload industrial unit operating 8,000 hours per year and consuming natural gas at controlled interstate prices. Table 7.19 shows the estimates that will be used throughout the remainder of the analysis. These estimates are believed to represent the actual costs incurred by users of gas turbines.

7.3.3 Impact of Environmental Control Costs

Section 7.2 of this analysis considered various water purification systems that could be used to provide water of the required quality and volume to control NO_x emissions from gas turbines. In this subsection, the costs of these water purification systems are converted into costs per kilowatt hour of net generation to determine the potential impact on the model plant applications described in Section 7.1. These costs are compared to the baseline estimates presented in the previous section to determine the relative impact in each application.

TABLE 7.19

BASELINE COST ESTIMATES
USED TO COMPARE THE
INCREMENTAL COST OF ENVIRONMENTAL CONTROLS

(Mills/Kwh)

Application	Baseline	Range
a. Standby Unit		
1. 80 hours	611.3	473.8 - 748.8
2. 200 hours	264.8	209.8 - 319.8
b. Industrial Application		
1. Single Shift (2000 hr)	43.8	23.7 - 47.5
2. Baseload (8000 hr)	32.5	12.5 - 33.4
c. Utility		
1. Peaking (500 hr)	85.5	69.8 - 100.5
2. Shoulder Peaking (2000 hr)	38.2	22.5 - 42.0

7.3.3.1 Parameter Values -- The most important determinants of the cost of using water injection to control NO_x emissions from gas turbines are:

(1) the capital costs of the water purification system;

(2) operating costs of the purification system;

(3) the water injection rate;

(4) cost of water transporting to the turbine site, if applicable; and

(5) any efficiency penalties associated with the injection of water.

Each of these factors are addressed below.

7.3.3.1.1 Capital and Operating Costs -- Section 7.2 considered alternative water purification systems. The systems selected for use in each application were designed to meet manufacturers' required water quality specification and flow rates. Sized for normal operating practice, the units should adequately meet the NO_x emission levels considered (i.e., 75 ppm, etc.) proposed by EPA. A summary of capital and operating costs of the systems selected for evaluation is provided in Table 7.20. Capital costs were based on manufacturers' estimates of installed costs and the cost of modifying the gas turbine to permit water injection. Operating costs included chemicals and other variable expenditures for proper operation of the purification system.

The same purification system would be adequate for either of the standby applications. If the turbine could be expected to operate more than 200 hours per year, a more expensive purification system could be purchased which would substantially reduce operating costs. Given normal operations

TABLE 7.20

CAPITAL AND OPERATING COST ASSOCIATED WITH WATER PURIFICATION EQUIPMENT

Application	Installed Capital[1]		Operating Costs	
	Total$	$/Kw	$/1000gal	$/yr
A. Standby unit				
1. 350 HP	1400	5.46	20.00	9.00
2. 1100 HP	1400	1.73	20.00	22.50
B. Industrial				
1. 4000 HP - typical	14000	4.78	3.00	175.00
2. 4000 HP - offshore	27000	9.21	11.00	9640.00
C. Utility				
1. 66000 KW	170000	2.58	2.90	7300.00

[1] Includes both the purchase price of the water purification equipment and the installation cost.

for standby units, however, it is unlikely that a more sophisticated system would prove to be economical. The cost estimates for the industrial applications are based on estimates provided by vendors of water treatment equipment.[71/] The offshore facility has considerably higher costs because of the need to desalinize the injection water. The utility estimates are based on a unit currently in service using water injection.[72/] The system is designed to meet state emission regulations of 75 ppm (ref. 15% oxygen) or the equivalent of regulations being considered by EPA.

7.3.3.1.2 Water Costs -- With the exception of the offshore drilling platform, all of the water purification systems are expected to utilize tap water supplied by local utilities. Normal water charges per 1,000 gallons range from $.20 to $.70.[73/] Table 7.21 gives water costs per 1,000 gallons for fifteen cities. The average cost of $.37 per 1,000 gallons was used to estimate water costs.

7.3.3.1.3 Transport Costs -- In a few isolated applications, such as remote pipeline compression stations, feed water for the injection system will not be available at the plant site and will have to be transported by truck and stored on site. The capital cost figures in Table 7.20 include the cost of water storage facilities. The costs of transporting the water have been estimated to be $.02 per gallon for a 50-mile, one-way trip and $.04 per gallon for a 100-mile trip.[74] It is highly unlikely that a source of potable water of sufficient quality would be more than 100 miles from any turbine location. One major interstate gas pipeline company has stated that water will be available at all of its compressor stations.[75]

TABLE 7.21

ESTIMATED AVERAGE COST PER THOUSAND GALLONS OF WATER IN 1974

City	Dec. '74 [1] $ per 1000 gals.	Out of City [2] Limits Market
Albuquerque, New Mexico	.233	100%
Atlanta, Georgia	.526	70%
Austin, Texas	.486	50%
Baltimore, Maryland	.220	Varies
Boston, Massachusetts	.646	None
Detroit, Michigan	.197	37-40%
El Paso, Texas	.240	$5.00 month
Kansas City, Missouri	.324	8%
Long Beach, California	.312	50%
Louisville, Kentucky	.553	None
Miami, Florida	.244	None
New York, New York	.704	None
Oklahoma City, Oklahoma	.282	34%
Seattle, Washington	.155	50%
Average	.366	

Source: Dallas Rate Survey, Journal AWWA, May '75.

[1] For users consuming more than 100,000 cubic feet per year.

[2] Potential increase in cost if located outside city limits.

7.3.3.1.4 Efficiency Penalty -- One manufacturer has estimated that water injection will result in a 2.9 percent heat rate penalty (BTU/kwh) and a 3.9 percent increase in the capacity of the unit.[76/] Westinghouse estimates that water injection will result in a penalty in the range of 1 to 1.5 percent with a 3 to 4 percent increase in the capacity of the unit at baseload.[77/] In evaluating the cost of the emission control system, a 1.5 percent heat rate penalty was assumed to occur in all applications. Enhancement of output capacity was only considered when the unit would be continuously operated at its rated capacity. In such instances the enhanced output of the unit might increase the annual output of the unit, thereby reducing the annualized cost per unit of output by spreading the fixed cost over a greater output. However, this will probably not occur in most instances because the purchaser will install a turbine which is slightly larger than required to meet maximum requirements. The only exception would be for baseload internal power generation and possibly compressor stations where turbines are operated at full load between major overhauls. The capacity expansion factor was incorporated by decreasing the installed costs per kilowatt by the percentage increase in the capacity of the unit. For example, if the cost per kilowatt was $120/kw, the cost after adjusting for the enhanced capacity would be $116.50 per kilowatt, which is a 3 percent decrease in the installed cost. All applications where this factor was considered to be important were analyzed with and without this adjustment to ensure that the range of possible control costs are presented.

7.3.3.2 Cost Estimates -- The model plant control cost estimates shown in Table 7.22 were used to calculate the percentage impact shown in Table 7.23 Eleven model plant applications were analyzed. The results in each case are explained below.

TABLE 7.22

PARAMETER ESTIMATES FOR MODEL PLANT COST EVALUATION

Parameters	Units	MODEL PLANTS Standby Units				MODEL PLANTS Industrial Application			MODEL PLANTS Utility				Offshore
		S-1	S-2	S-3	S-4	I-1	I-2	I-3	U-1	U-2	U-3	U-4	
Unit Size	Kw or HP	350 hp	350 hp	1,100 hp	1,100 hp	4,000hp	4,000hp	4,000hp	66,000 Kw	66,000 Kw	66,000Kw	66,000 Kw	4000 hp
Installed Cost	$/Kw	220	220	220	220	120	120	120	150	150	150	150	120
Heat Rate	BTU/Kwh	15,500	15,500	15,500	15,500	13,200	13,200	13,200	10,300	10,300	10,300	10,300	13,200
Fixed Capital Charge	%	20	20	20	20	20	20	20	20	20	20	20	20
Operating and Maintenance	%	1.0	1.0	1.0	1.0	5.0	5.0	5.0	1	1	1	1	5
Fuel Costs	$/MMBTU	2.18	2.18	2.18	2.18	2.18	2.18	2.18	2.18	2.18	2.18	2.18	2.18
Hours of Operation	Hr/year	80	200	80	200	2,000	8,000	8,000	500	500	2,000	8,000	8,000
Capital Costs of Purification Equipment	$/Kw	5.46	5.46	1.73	1.73	4.78	4.78	4.78	2.58	2.58	2.58	2.58	9.2
Operating Costs of Purification Equipment	$/1000 gals	20.00	20.00	20.00	20.00	3.00	3.00	3.00	2.90	2.90	2.90	2.90	11.00
Water Costs	$/1000 gals	.37	.37	.37	.37	.37	.37	.37	.37	.37	.37	.37	-----
Water Requirement	GPM	.1	.1	.25	.25	1.6	1.6	1.6	42	42	42	42	1.6
Water Transport Cost	¢/gal	---	---	----	----	----	----	2.0	----	----	---	-----	---
Efficiency Penalty	% BTU/Kwh	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Capacity Enhancement	% $/Kw	---	---	----	----	----	3.0	---	----	----	---	3.0	3.0
Baseline Costs	Mills/Kwh	611.5	264.8	611.5	264.8	45.8	32.5	32.5	180	85.5	38.2	26.4	82.5

TABLE 7.23

GAS TURBINE ENVIRONMENTAL CONTROL COSTS

(Mills/Kwh)

	Standby				Industrial			Utility				Offs Dril Plat
Water Injection Costs	S-1	S-2	S-3	S-4	I-1	I-2	I-3	U-1	U-2	U-3	U-4	
1) Annualize Fixed Costs	13.65	5.46	4.32	1.73	.48	.12	.12	2.58	1.03	.26	.06	
2) Operating Cost of Water Treatment	.46	.46	.37	.37	.10	.10	.10	.11	.11	.11	.11	
3) Water for Injection	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01	
4) Energy Penalty	.51	.51	.51	.51	.43	.43	.43	.34	.34	.34	.34	
5) Water Transport Costs	-----	----	----	----	---	---	.64	---	---	---	---	
6) Output Enhancement	-----	----	----	-----	---	-.09	-.09	---	---	---	-.11	
TOTAL	14.63	6.44	5.21	2.62	1.02	.57	1.21	3.04	1.49	.72	.41	
BASELINE COSTS	611.29	264.79	611.29	264.79	43.77	32.53	32.53	180.00	85.45	38.20	26.39	32
% IMPACT	2.39	2.43	.85	.99	2.32	1.75	3.71	1.69	1.75	1.88	1.56	2

7.3.3.2.1 Standby Units -- Four model plant applications were considered for standby power generating stations. The first two cases, S-1 and S-2, differ only in terms of the number of hours operated each year. The first unit operates approximately 80 hours per year and the second 200 hours per year. These units show the highest percentage impact in terms of the incremental costs per net kilowatt hour of power generation. The low number of hours operated each year tends to drive the cost of producing power up because fixed costs are spread over a relatively small base. The estimated impact in both cases was roughly 2.4 percent.

Cases S-3 and S-4 apply to 1,100 horsepower (hp) units operating the same number of hours, respectively, as the smaller 350 hp unit. This units can use exactly the same water purification system as the smaller unit. Consequently, while the costs of producing power, independent of the water injection system, are exactly the same, the percentage impact is much less, decreasing to less than 1.0 percent.

7.3.3.2.2 Industrial Units -- Four model plants were considered in the industrial category. The highest percentage impact was recorded in Case I-3, which represents a remote turbine application in an arid climate in which water must be transported fifty miles at a cost of $.02 per gallon. The impact in such cases, which includes water storage facilities, is approximately a 3.7 percent increase in the average cost of generating power. Since water injection results in a slight increase in the power output capacity of the unit, a credit was taken of .05 mills per kilowatt for the output enhancement. The unit is assumed to be baseload, operating 8,000 hours per year.

The remaining applications are as follows. Case I-1 represents a normal, single shift gas turbine application. The unit is operated 2,000 hours per year and is slightly oversized, therefore negating any benefits that might be derived from the improved output of the unit. Case I-2 is a baseload turbine operated 8,000 hours per year. A credit was taken, in this instance, for the improved capacity of the unit. All three units were sized at 4,000 hp.

7.3.3.2.3 Utility Applications -- Four model plant applications were considered for the utility industry. The first unit is operated 200 hours per year, the second 500 hours per year, third 2,000 hours per year and the fourth 8,000 hours. A credit for enhanced output was taken in the last case, since the unit is baseloaded. In all four cases, the percentage impact per kilowatt hour is less than 2 percent.

7.3.3.2.4 Offshore Drilling Platform -- Gas turbines are commonly used on offshore drilling platforms. Their compact size and low maintenance requirements favor their use for this type of application. Initially, it was thought that this case would evidence the highest percentage impact. The unit was assumed to use sea water to fuel the water purification system, resulting in a substantial increase in the capital and operating cost of the purification system. The installed cost of the combination desalination - purification system was $27,000 compared with $14,000 for the onshore application. Despite these high costs the fact that water would be available more than offset the higher costs associated with transporting water to the remote gas compressing station application. The total cost for water injection at the remote site was 1.21 mills/kwh compared with .92 mills/kwh for the offshore platform.

7.3.3.3 Sensitivity Analysis of Cost Impacts -- The cost estimates discussed above are partially dependent on assumptions underlying the cost model. Not every firm or utility will consider a fixed charge rate of 20 percent to be appropriate and, regardless of its availability, some firms will use controlled interstate natural gas in new turbine applications. Recognizing that the assumptions made in the baseline case will not hold in all cases, four of the parameters were given different values in an attempt to determine the sensitivity of the cost estimates. Fuel costs were reduced from $2.18/MMBTU to $.66/MMBTU, which is equivalent to the current city-gate price for natural gas. The fixed capital charge was given values of .25 and .15. Capital costs were increased to $270/kw for standby generators, $150/kw for industrial units, and $180/kw for the 66,000 kilowatt utility turbine. Finally, the output enhancement adjustment was dropped, eliminating the credit for the boost in power output associated with water injection. The resulting percentage impacts after making these changes are shown in Table 7.24.

The estimates are most sensitive to the change in fuel costs. In eight of the eleven model plant applications, reducing fuel costs from $2.18 to $.66/MMBTU resulted in the highest percentage impacts. The change was especially pronounced in the two cases where high water purification costs were anticipated. Lowering fuel costs in the remote site industrial application (I-3) accentuates the importance of water transport costs. In this case even though environmental control costs decrease, the decrease in the cost of power generation was even greater, thereby resulting in a net increase in the percentage impact.

TABLE 7.24

SENSITIVITY ANALYSIS OF ENVIRONMENTAL CONTROL COSTS

(% OF BASELINE COSTS)

	Standby				Industrial			Utility				Offshore Drilling Platforms
	S-1	S-2	S-3	S-4	I-1	I-2	I-3	U-1	U-2	U-3	U-4	
Baseline Cost Impact	2.39	2.43	.85	.99	2.32	1.75	3.71	1.69	1.75	1.88	1.56	2.83
Factor Analyzed												
1) Lower Fuel Cost $.66 vs $2.18/MMBTU	2.43	2.52	1.06	1.20	2.86	2.14	9.69	1.56	1.80	2.14	1.65	4.98
2) Fixed Capital Charge												
.25%	2.41	2.44	.84	.95	2.43	1.73	3.87	1.97	1.74	1.87	1.46	2.94
.15%	3.00	2.99	.87	1.04	2.20	1.76	3.78	2.12	1.75	1.90	1.67	2.71
3) Higher Capital Cost ($/Kw)												
. Standby $220-$270/Kw	1.97	2.03	.70	.82								
. Industrial $120-150/Kw					2.14	1.63	3.54					
. Utility $150-180/Kw								1.45	1.52	1.74	1.43	2.70
4) No adjustment for Enhanced Output						2.02					1.99	3.10
RANGE												
High	3.00	2.99	1.06	1.20	2.86	2.14	9.69	2.12	1.80	2.14	1.67	4.98
Low	1.97	2.03	.70	.82	2.14	1.63	3.54	1.45	1.52	1.74	1.43	2.70

The assumptions pertaining to capital costs and output enhancement, in general, did not cause any major changes in the percentage impacts. Increasing the cost of turbines relative to the cost of the emission control systems reduces the percentage impact, which was anticipated. The fixed charge rate generally has less of an impact, because it is applied to both the capital cost of turbines and the control equipment.

In summary, the cost of water injection control equipment for gas turbines should fall somewhere in the range of 2 to 14 mills per kilowatt hour for standby units, 6 to 1.2 mills per kilowatt hour for industrial applications, and 4 to 3.0 mills per kilowatt hour for utility gas turbines (Table 7.23). On a percentage basis, the potential impacts for standby units fall within a range of .7 to 3.0 percent. The range for industrial units is 1.6 to 9.7 percent and for utilities 1.4 to 2.2 percent (Table 7.24). Typical impacts will probably fall in the range of 2.0 percent regardless of the application, or rougnly 1 mill/kwh.

7.3.4 Impact of Environmental Control Costs for Regenerative and Combined Cycle Gas Turbines

No attempt has been made to provide detailed estimates of the control costs for regenerative and combined cycle gas turbines. The impacts, in absolute terms, are not expected to be much greater than for simple cycles moreover, the percentage impacts will be less given the higher cost per kilowatt of generating capacity for these units.

For combined cycles, the control costs should be the same as for simple cycles. If the costs of the associated boilers and steam generators are included, the percentage impact is certain to be less than for the simple cycle unit.

Manufacturers report that to control NO_x emissions[78] the regenerative unit will require a higher water injection rate than for similar size, simple cycle turbines. Therefore, the purification system will have to be slightly larger and cost more. For example, a 20 percent cost increase for a 4,000 horsepower unit would increase the cost by .03 mills per kilowatt hour for a unit operated 2,000 hours per year, the equivalent of a .1 percent increase in overall operating costs. If the water for the purification system also had to be transported, a 20 percent increase in the volume transported would increase the cost by .13 mill, or 4 percent. The maximum impact in this extreme case would be .5 percent over the estimate impact of 3.77 percent. Given these circumstances, it is believed that the cost impact for combined cycle and regenerative gas turbines will not be substantially greater than for the simple cycle, and the percentage impact should be in the same relative range. In the case of the combined cycle, if the entire turbine steam generator package was considered, the percentage impact would be smaller.

7.3.5 Alternative Control Levels

All alternative control levels less stringent than the 0.3 pounds per MMBTU under consideration by EPA would require less control. If the levels were increased to .5 pounds per MMBTU, or 125 ppm, the smaller standby units would probably not have to use water injection. The cost in this instance would be zero. Also, for the larger units the water injection rate would be less, reducing the amount of water consumed per kilowatt hour of net generation.

However, control costs would not decrease substantially because the water injection ratios would be much less important to the overall cost of control than the purity of the water and associated purification equipment. The cost per kwh for injection water is roughly .01 mill in every application. If the required water injection rate fell by 90 percent, it would only reduce the cost of the system by .009 mills/kwh, which is equal to .03 percent of the cost of operating the baseload utility application.

7.3.6 Summary

This section has attempted to provide estimates of the potential cost impacts of NO_x control equipment for gas turbines. The resulting estimates showed that, except for standby units, the total change in costs will probably fall in the range of .4 to 1.5 mills per kilowatt hour for turbines used in industrial and utility applications. Control costs for standby units, on a mills per kilowatt hour basis, were much higher, ranging from 2 to 14 mills per kilowatt hour, but this is primarily a function of the low number of operating hours. These costs are equivalent to about a 2.5 percent increase in operating costs for standby units and about 2 percent for industrial and utility applications.

7.4 ECONOMIC IMPACT

7.4.1 Impact on Manufacturers

The economic impact of imposing NO_x emission standards on the gas turbine industry will depend on the industry's ability to pass through the additional costs to consumers. Table 7.25 shows the expected increase in installed cost that will result from using water injection to control NO_x to 75 PPM. The percentage impact varies anywhere from .8 percent in the case of the 1100 hp standby unit to 7.1 percent for the unit requiring both desalination and purification equipment.

The major questions that must be resolved are the degree of competition between individual manufacturers and whether the pollution control cost will place the gas turbine industry at a competitive disadvantage with stationary reciprocating engines. Regarding the first issue, since all manufacturers would be forced to use basically the same purification system to meet the 75 ppm emission level, no manufacturer would gain a competitive advantage. Therefore, the degree of intra-industry competition should remain fairly constant.

The second and more important issue is the competitive position of the industry relative to stationary diesel generators. With respect to the largest segment of the industry (measured by capacity), utility peak load generators, diesel engines can not be produced in this size range. Therefore it can safely be assumed cross-price substitution impacts can be ignored. The same can not be said for standby generators and gas turbines used by the industrial sector. In both of these areas the competition between turbines and diesel engines is quite strong.

TABLE 7.25

IMPACT OF NO_x EMISSION CONTROL ON THE INSTALLED CAPITAL COST OF GAS TURBINES

Application	Installed Cost (1000$)		% Increase
	without controls	with controls	
A. Standby			
1. 350 HP	56.6	58.0	2.4
2. 1100 HP	177.9	179.3	0.8
B. Industrial			
1. 4000 HP - typical	352.8	366.8	4.0
2. 4000 HP - offshore	352.8	379.8	7.1
C. Utility			
1. 66,000 Kw	9900.0	10070.0	1.7

Historically, the price competition between the gas turbine and diesel engine industries has centered around first cost and fuel efficiency tradeoffs. The cost of reciprocating engines varies anywhere from 30 to 60 percent more than turbines, but they have the advantage of requiring less fuel per horsepower hour. While fuel efficiency depends on the type of operation, diesel engines are generally about 30 percent more efficient. Typical heat rates for turbines run in the range of 9700 BTU/hp-hr compared to 7,000 BTU/hp-hr for diesel engines.

The cost of generating power from these systems is plotted as a function of annual operating hours in Figure 7.2 for both the standby and industrial application. The assumptions underlining these curves are:

(1) the diesel is 30 percent more efficient that the turbine;

(2) the installed cost of the diesel is 30 percent more than the turbine;

(3) both are operated on distillate fuel oil priced at \$2.18/MMBTU; and

(4) operating and maintenance cost are essentially the same and both include the cost of environmental controls

Given these assumptions the curves show that the diesel is clearly superior, from a cost standpoint, to the turbine in industrial applications where the unit is operated 2,000 or more hours per year. The opposite is true in the case of the standby unit. The smallest cost differential for the industrial application is 5.1 mills/kwh, which is slightly less than five times greater than the estimated cost of the water injection emission control system of 1.02 mills/kwh.

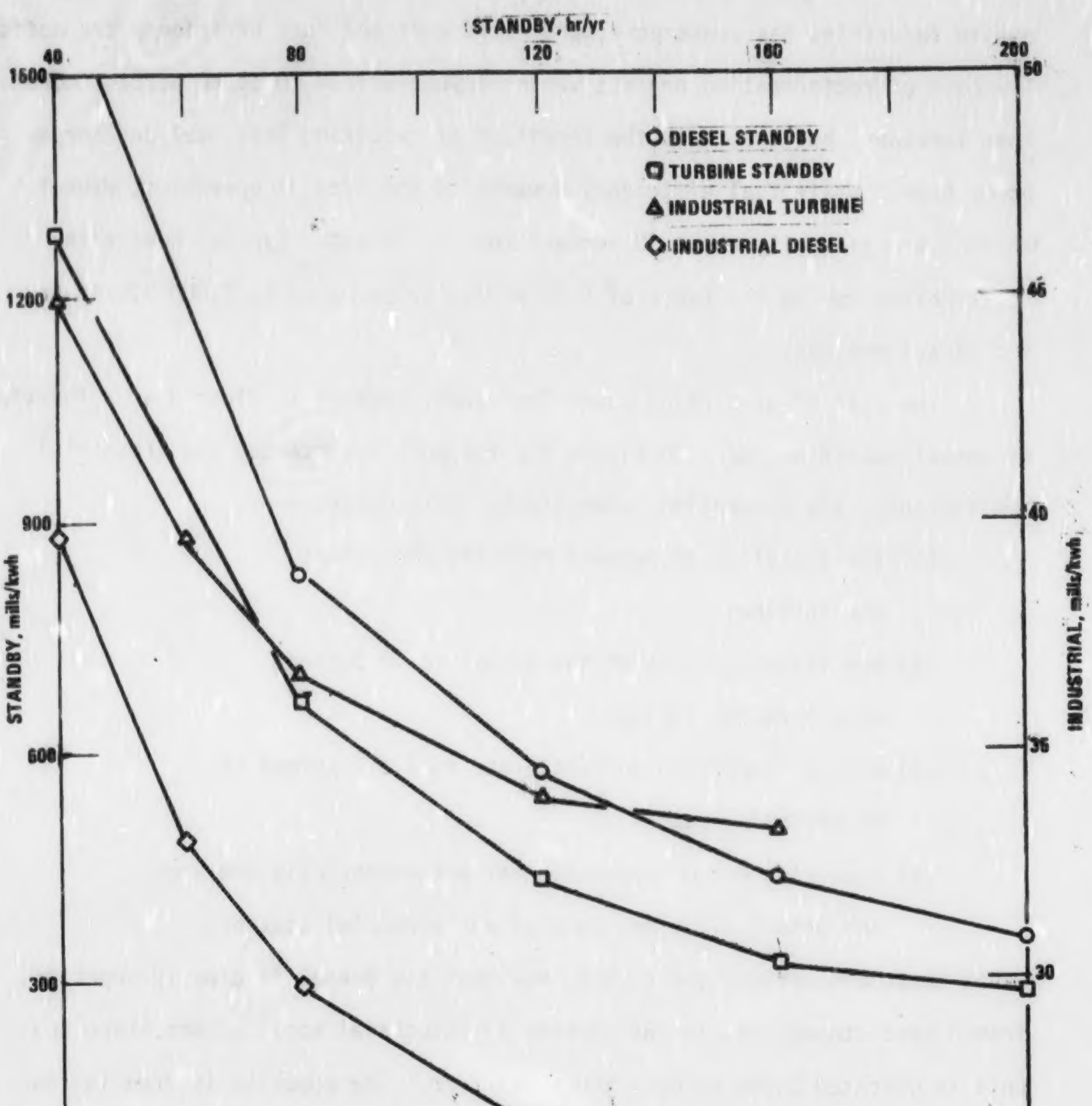

Figure 7.2. Comparative cost for gas turbines and diesel generators.

The difference is more pronounced in the case of the standby unit where the smallest cost differential is 52.2 mills/kwh compared to the incremental cost for emission controls of 2.62 to 6.44 mills/kwh (Table 7.23).

Another way of focusing on this issue is to solve for the number of hours of operation that would equate the two systems. The formula below can be used to determine the breakeven point:

$$\text{Mills/kwh} = \frac{(CRF)\ (K) + (OM)}{Hr/yr} + P_f\ (E)$$

where

CRF = capital recovery factor = 20%

K = installed cost of unit (diesel = 1.3 turbine)

OM = operating and maintenance expense (assume .05 K)

P_f = price of fuel in \$/MMBTU (assume \$2.18/MMBTU)

E = heat rate efficiency in MMBTU/kwh (assume turbine = 1.3 diesel)

To solve for the number of hours of operation that would equate the two systems, values must be assigned to the parameters. The assumed values for each of the parameters and applications are shown below.

Assumed Parameter Values

	Turbine	Diesel
Standby		
CRF (%)	.2	.2
K (\$/kwh)	220	286
OM (% K)	$.05K_T$	$.05K_D$
P_f (\$/MMBTU)	2.18	2.18
E (MMBTU/kwh)	.0155	.0109

Industrial

CRF (%)	.2	.2
K ($/kwh)	120	156
OM (% K)	$.05K_T$	$.05K_D$
P_f ($/MMBTU)	2.18	2.18
E (MMBTU/kwh)	.0132	.0092

The breakeven point for the standby unit is 1,650 hours per year; for the industrial application it is 1,034 hours per year. If these units were operated less hours per year, the turbine would have a competitive advantage, the opposite being true if the hours of operation exceeded these limits. It is important to note that standby units are seldom if ever operated more than 200 hours per year and industrial turbines are normally operated a minimum of 2,000 hours per year. If the sole determinant of market share between turbines and diesel engines were the cost of operating the unit, turbines would have total control of the standby market and diesels would control the industrial market. While cost is certainly important, it is not the sole determinant of the end user's selection criteria. Size, weight, and dependability in many cases are equally important and in some instances the overriding consideration.

The future growth and profitability of the gas turbine industry will depend upon many factors other than pollution control costs. The most important factor is probably the price of fuel oil. In the last section the cost of controlling NO_x through the use of water injection was shown to fall in the range of .5 to 1.2 mills/kwh. This cost impact is approximately equivalent to a 1.7 to 4.2 percent increase in the price of distillate fuel oil. If fuel oil prices continue to increase as rapidly as they have over the past few years, in the absence of substantial fuel efficiency gains any competitive advantage once held by the turbine industry over diesel will be rapidly eliminated.

On the basis of the preceeding discussion it would be difficult to make a case for NO_x controls having a substantial economic impact on the manufacturers of gas turbines. First, based on existing emission rates all manufacturers will have to employ the use of wet control to meet the 75 PPM NO_x standard. This would imply that intra-industry competition should remain constant. Second, in comparing turbines with diesel generators first cost and fuel efficiency tradeoffs are important but not necessarily the only determinants of choice. If price alone was the primary determinant, the market would be considerably more segmented than it is. Third, the most important market based on capacity for gas turbines is the electric utility industry; diesel generators are not produced in this size range. Therefore the imposition of NO_x controls should not impact on either the degree of intra- or inter-industry competition. Since the market positions of manufacturers in the industry will not be affected by the imposition of NO_x standards, the manufacturers should be in a fairly strong position to pass additional costs on to end-users.

7.4.2 Impact on Consumers

7.4.2.1 Utilities -- The impact of controlling NO_x emissions from gas turbines on end-users should be primarily a function of the number of operating hours and the user's dependency on turbines for power generation. The most important market for gas turbines is the electric utility industry, where turbines account for approximately 8.25 percent (39,300 MW) of total generating capacity and 1.6 percent of net power generation.[79/] Total electricity production by the electric utility industry over the twelve months ending in June, 1975 was 1,886 x 10^9 kwh.[80/] The average cost of generating power was 27.65 mills per kwh.[81/]

If all the electric utility industry's turbines were using wet controls, the percentage impact on the cost of generating power would have been .09 percent, based on an average cost of 1.5 Mills/kwh for wet controls and 1.6 percent of net power generation attributable to gas turbines. To place this figure in perspective, a change in the cost of generating power of this magnitude would be roughly equivalent to a .5 percent increase in the price of coal. Since the emission level under consideration here, if adopted, would only require control on new gas turbines, the full .09 percent increase in the cost of generating power will occur until all existing turbines have been replaced. Gas turbines have an extremely long operating life (in excess of 20 years); therefore, the total impact will not be felt until somewhere in the 1995 timeframe.

Also, since the utility industry is a franchised monopoly, they should be allowed to recover any costs associated with water injection to control NO_x emissions from gas turbines through rate adjustments. This has been true for other pollution control expenditures and there is no reason to assume differently in this instance.

7.4.2.2 Industry -- The impact on the industrial sector is a function of each industry's dependence on electricity and the degree that it relies on internal power generation. Table 7.26 shows the amount of energy consumed per dollar of output in the seven most energy intensive industries. Also shown is the percentage of total energy consumed in the form of electricity and the percent derived from internal power generation. The cost of wet controls on gas turbines can be converted into a cost per million BTU simply by dividing the incremental cost for the NO_x emission control system (1.2Mills/kwh) by the amount of energy (.0132 MMBTU) consumed per kwh of power generation. Therefore, the incremental

POTENTIAL IMPACT OF NO_x CONTROLS ON OUTPUT PRICES FOR THE SEVEN MOST ENERGY INTENSIVE INDUSTRIES

Industry	Energy/Output coefficient[1] (MMBTU/$ Output)	Share of Electrical Energy[2] (%)	% Self Generated[3] (%)	Equivalent Increase in Price of Fuel $/MMBTU	Impact on Product Prices (%)
Steel	.140	9.6	16.6	.09	.020
Chemicals	.072	15.8	12.9	.09	.013
Petrochemicals	.144	18.0	29.2	.09	.068
Petroleum	.050	9.0	14.1	.09	.006
Paper	.139	12.0	40.0	.09	.060
Aluminum	.140	53.7	11.7	.09	.079
Cement	.267	14.7	4.6	.09	.016

[1] "Impact of Proposed Energy Deregulation/Tax Program on Five Energy Intensive Industries," March 16, 1975, EEA

[2] "Energy Consumption in the Industrial Sector: Data Sources for 1974 Consumption, " January 13, 1975, EEA, pg 3

[3] "Annual Survey of Manufacturers, 1974: Fuels and Electric Energy Consumed," U.S. Doc., Bureau of Census 1974 (AS)-4.1

cost per kwh for NO_x controls on gas turbines would be equivalent to approximately \$.09/MMBTU. By multiphying the energy/output coefficient for each of these industries by the share of electricity, the percentage of electricity which is self generated and the fuel price equivalent of the cost of wet controls, it is possible to approximately indicate the potential impact on product prices for the highly energy intensive industries. The price rises most in the case of aluminum where, if all internal power was generated from gas turbines using water injection, the imposition of a 75 ppm NO_x standard would lead to a .079 percent increase in product prices. These estimates are certain to overstate the price impact, because many sources of power include hydro, steam electric and diesel engines. The figures in Table 7.26 indicate that even under worst case assumptions the impact on industrial end-users will be quite small.

7.4.2.3 Standby - The potential impact on users of standby turbines is also believed to be quite small. Since the units are only operated once or twice a year and the capital cost is expected to rise by .8 to 2.4 percent, the economic impact on end-users of standby units will be quite small.

7.4.3 Total Capital Cost, Annualized Cost and Energy Penalty

The total cost to the nation of imposing a 75 ppm NO_x standard on the stationary gas turbine industry has been calculated on the basis of expected sales for the period 1977 to 1982. Sales projections were based on Section 3.1 of this report. Costs are shown both in terms of total capital outlays and annualized cost as a function of assumed hours of operation.

7.4.3.1 Projections of New Capacity

Table 7.27 shows turbine capacity on-line in 1974 and projections

TABLE 7.27

SALES PROJECTIONS AND INCREMENTAL CAPACITY1/

Application	Total Capacity			Incremental Sales
	1974	1980	1985	7/77-6/82
Utilities (MW)				
EEA (High)	39,300	57,000	77,000	20,972
TBS (Low)	39,300	50,700	64,000	13,676
Oil and Gas Ind. (10^3HP)				
A-D (High)	6,831	10,755	14,904	4,509
BAU (Low)	6,831	8,242	9,423	1,429
Standby (10^3HP)	1,257	2,450	3,952	1,509
Private Electric (MW)				
Power Generation				
a. oil/gas Ind.				
A-D (High)	460	1,003	1,164	419
BAU (Low)	460	500	536	41
b. other industry	1,900	2,695	3,600	943
Other applications (10^3 HP)	1,147	1,627	2,175	570

1/ Sales projections based on data presented in Chapter 3.

for 1980 to 1985. To conform to the economic impact guidelines these estimates have been adjusted to a five year period beginning in July of 1977 and ending in June of 1982 Incremental sales for this period are shown in the last column. All cost estimates and energy penalties are keyed to these sales projections.

7.4.3.2 Capital and Annualized Total Costs - Table 7.28 presents estimates of the total capital outlay required to equip gas turbines with water injection systems to control NO_x emissions. The total capital cost for the years 1977 - 1982 should fall somewhere in the range of $53 to $85 million dollars. The total annualized cost will scale up from zero in 1976 to between $19 and $40 by mid-year 1982. Assuming a linear growth rate the total cost over the five year period should fall somewhere in the range of $60 to $120 million depending on sales volumes.

7.4.3.3 Energy Penalty - The energy impact of these regulations will depend on the efficiency loss associated with using water injection systems, the heat rate (Btu/kwh), hours of operation and total capacity expected to come on-line. Table 7.29 provides two projections that vary only in incremental sales volumes. The energy impact is projected to be between 1,500 and 3,800 barrels of oil per day by mid-year 1982, compared to a total nationwide consumption of 18 million barrels per day, or .02 percent. This increase in fuel oil consumption will gradually be reduced to zero as emission levels are met by the application of the dry control techniques discussed in Chapter 4. Most companies anticipate that dry controls will be applied within five years.

TABLE 28

TOTAL CAPITAL AND ANNUALIZED COST

	1977-1982 Incremental Sales[1] (MW)		Water Injection Capital Costs ($/Kw)	Total Capital Cost 1977-1982 (10^6 $)		Assumed Average Annual Usage (Hr/year)	Annualized cost per Kwh Mills/Kwh	Total Annual Cost in 1983 (10^6 $)	
	High	Low		High	Low				
Utilities	20,972	13,626	2.58	54.1	35.2	500	1.5	15.7	10.2
Oil and Gas	3,382	1,072	4.79	16.2	5.1	8,000		16.2	5.1
Standby	1,192		5.46	5.8		80	5.2	.5	
Internal Power									
a. oil/gas	419	41	4.78	2.0	.2	8,000	1.2	4.0	.4
b. other industries	943		4.78	4.5		2,000	1.0	1.9	
Other application	428		4.78	2.0		2,000	1.0	.9	
Total				84.6	52.8			39.2	19.0

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[1] Horsepower has been converted to kilowatts for ease of exposition.

TABLE 7.29

ENERGY PENALTY

Application	Incremental Sales (MW) High	Incremental Sales (MW) Low	Assumed Average Annual Usage (Hours/year)	Heat Rate (BTU/Kwh)	Efficiency Penalty * (%)	Energy Impact (10^9 BTU) High	Energy Impact (10^9 BTU) Low
Utilities	20,972	13,626	500	10,300	1.5	1,620	1,052
Oil and Gas	3,382	1,072	8,000	13,200	1.5	5,357	1,698
Standby	1,132		80	15,500	1.5	21	
Internal Power							
a. oil/gas	419	41	8,000	13,200	1.5	664	65
b. other industries	943		2,000	13,200	1.5	373	
Other application	428		2,000	13,200	1.5	169	
Total						8,204	3,378
Equivalent barrels of oil per day						3,746	1,542

* 0.5 water to fuel

7.4.4 Impact on Balance of Trade

The export market for gas turbines has been growing quite rapidly over the past five years. The imposition of NO_x standards should not affect this market potential because the controls on NO_x will be external to the basic operation of gas turbines. Unless the foreign buyer required NO_x controls, turbines available to satisfy foreign demand will be essentially the same units that are currently available. If dry controls are developed it is likely that the costs to meet the 75 PPM NO_x limitation will be no greater and probably less, both in terms of first costs and operating costs to the consumer than the cost of water injection. Any research and development costs for dry control will probably be amortized over total sales, including foreign sales which would affect the purchase price to foreign buyers. Given the status of research on dry controls it is impossible to estimate what it will cost to develop dry controls, which also make it impossible to estimate the potential secondary impact on export markets.

7.4.5 Impact of Alternative Level of Control

The alternative levels of control which will be considered here are 100 and 125 PPM (Ref 15% oxygen). If these standards were adopted, a significant percentage of the gas turbine industry would not have to use water injection systems to control NO_x. Based on the results of Section 7.2 only the large turbines in the range of 20,000 MW and above would be required to use water injection systems. Table 7.30 shows the distribution of sales by size range and manufacturer in this size range.

At present, for those machines which require water injection, the only difference among the alternate levels of control is the amount of water which must be injected into the turbine. As discussed earlier, the costs of water injection are not that sensitive to the amount of water injected over relatively small differences in water flow rates. The decrease in the flow rate should result in a 10-15% reduction in the capital cost of the water treatment facilities. This capital cost decrease would reduce overall cost impacts from 1.6-1.9% under the 75 PPM level to 1.5 to 1.8%. This would reduce annualized cost to utilities by $.7 to $1.1 million per year.

7.4.6 Cost Effectiveness of Control System

Table 7.31 shows the cost-effectiveness ratios measured in dollars per pound in NO_x emissions. The range for utility and industrial applications is from $.07/lb to $.25/lb of NO_x emission reduction. The cost-effectiveness is considerably lower for standby units, ranging from $.87/lb to $4.88/lb. Standby units have considerably lower NO_x emissions characteristics which, when combined with the high control costs, lead to a much higher cost per pound decrease in NO_x emissions.

TABLE 7.30
1974 Sales Volumes[1/] and Market Shares

Manufacturer	Size Range 20115-33526 HP	 33527-67051 HP	 67052-132102 HP	 >100MW 134,103 HP	1974 Sales Volumes HP	Market Share (%)
General Electric	196.6	103.4	3509.0		3809.0	34.4
TP & M		118.1	903.4	3892.0	4913.5	44.3
Westinghouse		145.0	870.2	731.6	1746.8	15.8
Turbodyne			484.4		484.4	4.4
Others		132.1			132.1	1.2
Total	196.6	498.6	5767.0	4623.6	11085.8	100.0*

Note HP - (000)

* Failure to sum % 100 percent due to rounding error.

1/ Sales volumes based on results of Section 3.

Table 7.31

COST-EFFECTIVENESS COMPARISON

Application	Size	Hrs/Yr	Emission Rate ($lbs NO_x/hr$)	Decrease in Emissions/kwh ($lbs NO_x/kwh$	Cost of Control (mills/kwh)	Cost-Effectiveness ($\$/lbs NO_x$)	Capital Cost ($)
A. Standby							
S-1	350hp	80	2.0	.0030	14.63	2.877	1,400
S-2	350hp	200	2.0	.0030	6.44	2.147	1,400
S-3	1,100hp	80	6.4	.0030	5.21	1.737	1,400
S-4	1,100hp	200	6.4	.0030	2.62	.873	1,400
B. Industrial							
I-1	4,000hp	2,000	24.2	.0041	1.02	.249	14,000
I-2	4,000hp	8,000	24.2	.0041	.57	.139	14,000
I-3	4,000hp	8,000	24.2	.0041	1.21	.295	14,000
C. Utility							
U-1	66MW	200	596.0	.0059	3.04	.520	170,000
U-2	66MW	500	596.0	.0059	1.49	.253	170,000
U-3	66MW	2,000	596.0	.0059	.72	.122	170,000
U-4	66MW	8,000	596.0	.0059	.41	.069	170,000
D. Offshore	4,000hp	8,000	24.2	.0041	.92	.224	27,000

7.5 POTENTIAL SOCIO-ECONOMIC AND INFLATIONARY IMPACTS

The Administrator has determined that for any action resulting in a greater than 5 percent price increase, an inflation impact statement should be prepared. Gas turbines for offshore platforms will cost 7.7 percent more when equipped with air pollution controls. Yet a vast majority of the offshore platforms will utilize turbines smaller than 10,000 hp and will thus be exempt from the regulation for five years under the recommended standard. Nevertheless, it is likely that some of the larger new platforms will require turbines of greater than 10,000 hp. However, these will constitute such a small percentage of the overall market that the net price effect will be well below five percent. Therefore, a formal inflation impact statement is not required.

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8. RATIONALE

8.1 SELECTION OF SOURCE FOR CONTROL

As described in chapter 3, stationary gas turbines are sources of particulate, NO_x, SO_2, CO, and HC emissions. NO_x emissions, however, are of more concern than emissions of these other pollutants for two reasons. First, NO_x is emitted in greater quantities from stationary gas turbines than are these other pollutants. Second, EPA has assigned a high priority to development of standards of performance limiting NO_x emissions. Assuming existing levels of emission controls, national NO_x emissions from stationary sources are projected to increase by about 65 percent by 1985. Applying best technology to all new sources would reduce this increase to about 25 percent, but would not prevent it from occurring. This unavoidable increase in NO_x emissions is attributable largely to the fact that current NO_x emission control techniques are essentially design modifications, not add-on equipment or operation changes. In addition, few NO_x emission control techniques can achieve large (i.e., in the range of 90 percent) reductions in NO_x emissions. Consequently, EPA has assigned a high priority to the development of standards of performance for major NO_x emission sources wherever significant reductions in NO_x emissions can be achieved.

Several studies sponsored by EPA have classified stationary gas turbines as major controllable sources of NO_x emissions. One study conducted by the Aerotherm Division of Acurex Corporation in 1975 estimated that oil-fired and gas-fired stationary gas turbines accounted for 2.5 percent of the total NO_x emissions from stationary sources in the U.S. in 1972. This same study ranked gas-fired turbines as sixteenth and oil-fired gas turbines as twenty-third in a priority listing, by equipment type and firing type, of 137 controllable stationary sources of NO_x emissions.

In another study The Research Corporation of New England (TRC) determined the impact which standards of performance would have on nationwide emissions of particulates, NO_x, SO_2, HC, and CO from stationary sources. Sources were ranked according to the impact, in tons/year of pollutant, which a standard promulgated in 1975 would have on emissions in 1985. This ranking placed gas turbines first on a list of 40 stationary NO_x emission sources.

In a subsequent study, Argonne National Laboratory (ANL) expanded the results of the TRC study to develop a priority listing of sources of particulates, NO_x, SO_2, HC, and CO, with CO not considered a pollutant for control by itself. In developing this listing, source screening factors were used which considered items such as:

- type, cost, and availability of control technology
- emission measurement methods and applicability
- enforceability of regulations
- source location and typical source size
- energy impact
- impact on scarce resources
- other environmental media constraints

The priority listing, which ranked 237 source-pollutant combinations in the order in which standards of performance should be developed, ranked utility gas turbines fifth and pipeline gas turbines twelfth with NO_x being the pollutant identified for control.

In 1974, 90 percent of all domestic stationary gas turbine capacity was sold to the electric utility market, primarily for use as peaking units. It is expected that this large percentage of sales to utilities will continue in the future due to the many advantages of gas turbines as peaking units. Stationary gas turbines are characterized by low capital cost, ease of installation, minimal maintenance requirements, and low physical profiles (low buildings and short stacks), all of which make gas turbines attractive for peaking units. In addition, turbines can be instrumented for remote operation, have very low visible emissions, and when properly muffled are quiet in operation. As a result, turbine peaking units are often located in large urban centers where power demands are greatest and pollution problems are often most severe.

Stationary gas turbines, therefore, are significant contributors to total nationwide emissions of NO_x. They are ranked high on the various listings of sources for which standards of performance should be developed. In addition, the expanding market for gas turbines coupled with the probability that many turbines will be installed near large urban centers underscores the necessity of developing standards of performance for stationary gas turbines. Consequently, stationary gas turbines are selected for development of standards of performance.

8.2 SELECTION OF POLLUTANTS

The pollutants emitted from stationary gas turbines are particulates, NO_x, SO_2, CO and HC. The nature and amounts of each of these pollutant emissions from gas turbines vary a great deal and have been discussed in chapters 3 and 4.

As discussed above, stationary gas turbines are a significant source of NO_x emissions, ranking first out of a list of 40 sources of NO_x

emissions according to the TRC report and fifth out of 237 source-pollutant combinations for utility turbine NO_x emissions and twelfth out of 237 source-pollutant combinations for pipeline turbine NO_x emissions according to the ANL report. Combustor modifications (dry control) and water injection (wet control), as discussed in chapters 4 and 7, are demonstrated techniques for reducing NO_x emissions at reasonable cost and, as discussed in chapter 6, depending on the specific emission level selected, could reduce NO_x emissions by as much as 190,000 tons per year in 1982. This is a significant decrease in total nationwide NO_x emissions. For these reasons, NO_x emissions from stationary gas turbines are selected for control by standards of performance.

SO_2 emissions from stationary gas turbines depend on the sulfur content of the fuel since nearly 100 percent of the sulfur in the fuel is converted to SO_2 during combustion. The TRC report ranked gas turbines 8 out of 41 major contributors of SO_2 emissions. Due to the high volumes of exhaust gases, as discussed in chapter 7, the cost of flue gas desulfurization (FGD) to control SO_2 emissions from stationary gas turbines is considered unreasonable. Selection of low sulfur fuels, however, is considered reasonable. Control of SO_2 emissions, therefore, requires combustion of low sulfur fuels rather than the application of FGD.

Since stationary gas turbines are a major source of SO_2 emissions and selection of low sulfur fuels is considered an economically feasible control technique, SO_2 emissions from stationary gas turbines are selected for control by standards of performance.

Hydrocarbon and carbon monoxide emissions from stationary gas turbines operating at full load are relatively low because the higher the percentage of full load at which a turbine operates, the more efficient the combustion of the fuel and the lower the emissions of HC and CO. Gas turbines normally operate at 80 to 100 percent of full load with HC emissions averaging less than 20 ppm and CO emissions averaging less than 150 ppm at 15 percent O_2. On occasion utility gas turbines may operate at 20-30 percent of load when in the "spinning reserve" mode. As shown by the data presented in Appendix C, at 10-20 percent of full load, HC emissions from large utility gas turbines are usually less than 50 ppm and CO emissions are usually less than 500 ppm at 15 percent O_2. Information submitted to EPA by the Edison Electric Institute,[1] however, indicates that utility gas turbines very seldom operate in the "spinning reserve" mode due to economic considerations. This conclusion was also confirmed in telephone conversations between utility and EPA personnel.

HC emissions from gas turbines were not included in the rankings of either the TRC or [illegible] reports. CO emissions from gas turbines were ranked sixth out of 42 contributors of CO in the TRC report but were not ranked in the more comprehensive ANL report.

Gas turbines will be operated as efficiently as possible to conserve fuel which at the same time will reduce HC and CO emissions; therefore, HC and CO emissions from stationary gas turbines are not selected for control by standards of performance.

Particulate emissions from stationary gas turbines are minimal as shown by the TRC report which ranks particulate emissions from gas turbines 111 out of 113 particulate emitters, and the ANL report which ranks utility and pipeline gas turbines 228 and 229, respectively, out of 237 possible source-pollutant combinations for particulate emissions. Consequently, particulate emissions from stationary gas turbines are not selected for control by standards of performance.

8.3 SELECTION OF AFFECTED FACILITIES

Stationary gas turbines are produced in three different configurations: simple cycle, regenerative cycle, and combined cycle. All of these configurations emit NO_x and SO_2, and all can be controlled for NO_x emissions by water injection or dry controls and for SO_2 by firing low sulfur fuels. Consequently, simple cycle turbines, regenerative cycle turbines and the gas turbine portion of combined cycle plants are selected as affected facilities for standards of performance limiting NO_x and SO_2 emissions.

Gas turbines can burn either liquid or gaseous fuels. The dry and wet control techniques described in chapter 4 for the control of NO_x emissions can be applied to gas turbines regardless of the type of fuel burned. Similarly, the firing of low sulfur fuel for the control of SO_2 emissions can be applied to gas turbines regardless of the type of fuel burned. Consequently, gas turbines burning all types of fuels are selected as affected facilities for standards of performance limiting NO_x and SO_2 emissions.

As discussed earlier, 90 percent of all stationary gas turbine capacity was sold to the electric utility industry in 1974 and this trend is expected to continue. In most cases these gas turbines were large units of 10,000 hp or more. Most of the remaining 10 percent of the gas turbine capacity sold in 1974 consisted of units in the range of 1000-10,000 hp. Thus, the contribution of small gas turbines of less than 1000 hp to national NO_x emissions is negligible. For many applications up to about 10,000 hp, stationary gas turbines compete with stationary internal combustion engines. A standard of performance on one of these industries and not the other, therefore, would tend to give the non-regulated industry a competitive advantage to some extent.

Currently, standards of performance are being developed for stationary internal combustion engines. Although relatively few internal combustion engines of greater than 1000 hp are produced, these engines are responsible for 75 percent of the total NO_x emissions from stationary internal combustion engines. Under 1000 hp, however, the number of internal combustion engines produced increases tremendously and enforcement of standards of performance would not be feasible in the absence of a certification program similar to that for automobiles. Since the Clean Air Act does not permit standards of performance to be enforced by a certification program, a lower size cutoff of 1000 hp for standards of performance for stationary internal combustion engines is considered appropriate. Consequently, to be consistent a lower size cutoff of 10.7 gigajoules per hour heat input (about 1000 hp) is selected for standards of performance for stationary gas turbines. Below this cutoff the standards limiting NO_x and SO_2 emissions would not apply.

Some gas turbines are operated as a mechanical or electrical power source only when the primary power source for a facility has been rendered inoperable by an emergency situation. This type of turbine operates infrequently, usually only for checkout and maintenance; therefore, it contributes only a very small amount to total nationwide NO_x emissions emitted by all gas turbines. Since this type of turbine is operated infrequently, the owner of such a turbine would probably choose, for economic reasons, to store water for injection rather than installing a water treatment unit on site. If the emergency situation was of such duration that the turbine used the entire supply of injection water before it could be replenished, then the turbine would have to shut down, leaving the facility the turbine served with no source of power. This situation could possibly be dangerous in some cases, such as where turbines are used to supply emergency power to hospitals. There also could be operational problems with the water injection system due to the long periods of non-operation. Consequently, emergency standby gas turbines are exempted from standards of performance limiting NO_x emissions.

Gas turbines could possibly contribute to the creation of ice fog, which consists of small (mean diameter 3 to 7 microns) ice crystals which are nucleated by airborne particulate. Ice fog occurs at temperatures below -28°C and is a serious problem in only a small portion of the United States, primarily Alaska. Ice fog severely restricts visibility and, since the crystals are long-lived, can plague auto and air traffic for extended periods. The actual impact of water or steam injection on the formation of ice fog is unknown; however, water or steam injection will increase the moisture content of the exhaust gas discharged by gas turbines. Since ice fog occurs only in a small portion of the United States and only under special weather conditions, the impact on air quality due to increased NO_x caused by exempting gas turbines creating ice fog would be minimal. Therefore, gas turbines using water or steam injection for control of NO_x emissions are exempted from the standards limiting NO_x emissions when ice fog created by the turbine is deemed by the owner or operator of the turbine to be a traffic hazard.

Stationary gas turbines are sometimes used by the military in combat-type situations. The main advantage of these turbines is their mobility, which would be considerably restricted by a water injection system consisting of either water treatment equipment or a water storage vessel. Restriction of the mobility of these turbines could have an adverse effect on national defense; therefore, any military combat-type gas turbine for use in other than a garrison facility is exempt from the standards limiting NO_x emissions.

The possibility of exempting some gas turbines from the standard limiting SO_2 emissions was also examined. Except for exempting all turbines of less than 10.7 gigajoules per hour heat input (about 1000 hp), no exemptions from the SO_2 emission limit were considered necessary.

8.4 SELECTION OF THE BEST SYSTEM OF EMISSION REDUCTION

As discussed in chapter 4, there are three possible control techniques for reducing NO_x emissions from stationary gas turbines: wet controls, dry controls, and catalytic exhaust gas cleanup. Wet controls involve the injection of water or steam into the combustion reaction to reduce peak flame temperatures, thereby reducing NO_x formation. Wet control techniques have been demonstrated on many large gas turbines (greater than 10,000 hp) used in utility and industrial applications, and these installations have had good reliability over long periods of operation. Wet controls, however, have not been demonstrated on small production gas turbines (less than 10,000 hp), although the effectiveness of these techniques for small gas turbines has been evaluated in laboratory and combustor rig tests. Thus, wet controls can be applied immediately to large stationary gas turbines, but manufacturers estimate that at least three years are required to incorporate and demonstrate wet control techniques on small production gas turbines.

Dry controls consist of operational or design modifications which govern combustion conditions to reduce NO_x formation. Although dry controls have been tested in laboratory and combustor rig tests, manufacturers estimate that up to five years is required for further development, design, test, and incorporation of dry controls on large and small stationary gas turbines. Catalytic exhaust gas cleanup consists of NO_x reduction by ammonia in the presence of a catalyst. While laboratory tests are very promising, this technique is not demonstrated. Consequently, only wet controls and dry controls are considered as viable alternatives for development of standards of performance.

The NO_x emission reduction achievable with these two alternatives clearly favors the development of standards of performance based on wet controls. Reductions in NO_x emissions of more than 70 percent have been demonstrated using wet controls. Dry controls, however, have demonstrated NO_x emission reductions of only about 30 percent. Laboratory tests have indicated that some dry control techniques have the potential of achieving much greater NO_x emission reductions, but these techniques need considerable development before they can be considered demonstrated.

Standards of performance based on wet controls would reduce national NO_x emissions by about 190,000 tons per year in 1982. In contrast, standards of performance based on dry controls would have no impact on national NO_x emissions in 1982, due to the necessity of allowing a five-year delay to incorporate dry controls on gas turbines. By 1987, standards based on wet controls would reduce national NO_x emissions by about 400,000 tons per year, whereas standards based on dry controls would reduce NO_x emissions by only about 90,000 tons per year. Thus, standards of performance based on wet controls would have a much greater impact on national NO_x emissions than standards based on dry controls.

Uncontrolled ambient NO_x levels near stationary gas turbines are typically well below the National Ambient Air Quality Standard of 100 $\mu g/m^3$ due primarily to the high temperature of the exhaust gases and resulting rapid plume rise. Typically, uncontrolled annual average ambient NO_x concentration levels from dispersion model studies of stationary gas turbines range from below 1 to about 14 $\mu g/m^3$. One model plant calculation, however, yielded an uncontrolled ambient NO_x level of about 50 $\mu g/m^3$. Standards of performance based on wet controls would reduce this ambient concentration to about 10 $\mu g/m^3$ while standards based on dry controls would only reduce this ambient NO_x concentration level to about 30 $\mu g/m^3$. Where ambient NO_x concentration levels near stationary gas turbines would be significant, therefore, standards of performance based on wet controls would be more effective in reducing ambient NO_x concentration levels than standards of performance based on dry controls.

The water pollution impact of standards based on wet controls would be minimal. Water needed for wet controls may be treated by the same processes used to treat steam boiler make-up water. The quality of the wastewater from this treatment is essentially the same as the influent water except that the concentration of total dissolved solids in the effluent stream is 3 to 4 times that of the influent. In most cases, the effluent may be sewered directly or returned to the river supplying the water. Where this is not possible, the effluent may be discharged to an evaporation pond. Consequently, the water pollution impact of standards based on wet controls would be minimal.

The quantity of water required by a stationary gas turbine using wet controls is relatively small. The upper limit water-to-fuel ratio of about 1:1 requires only about 5 percent of the quantity of water consumed by a comparable size steam boiler using cooling towers. A water treatment system for five 28 MW stationary gas turbines operating 10 hours per day using a water-to-fuel ratio of 1:1, for example, would treat 125,000 gallons of water and reject about 25,000 gallons of wastewater per day. A steam boiler of comparable size with cooling towers would consume about 20 times as much water. In fact, the usage rate of water for wet controls is small enough that the unlikely prospect of having to truck water 50 miles was determined to be economically reasonable as discussed below. Standards based on dry controls, however, would have no impact on water pollution or water supplies.

Standards based on wet controls would have a negligible solid waste impact. When it is not possible to sewer the wastewater effluent directly or to return it to the source from which the water was drawn, the use of an evaporation pond will result in the slow buildup of solids in the pond. These solids can be periodically collected and disposed of in landfills. Standards based on dry controls, however, would have no solid waste impact.

There would be no adverse noise impact resulting from standards based on either wet or dry controls.

The potential energy impact of standards based on wet controls is small. Standards based on wet controls could increase the fuel consumption of a gas turbine by as much as 5 percent, depending on the rate of water injection required to comply with the standard. Few turbines will require the high water injection rates (about 1:1 water-to-fuel ratios) which result in a 5 percent fuel penalty. Assuming that all stationary gas turbines subject to compliance with standards would require a 1:1 water-to-fuel ratio, the fifth-year energy impact on large stationary gas turbines would be about 5500 barrels of fuel oil per day in 1982. The fifth-year energy impact on small stationary gas turbines would be about 7000 barrels per day in 1987, as a result of the delayed effective date of the proposed standards on small turbines. Each increase represents less than a 0.03 percent increase in estimated oil consumption in the United States in 1982 and 1987. It should also be recognized that these energy impacts are based on assumptions which yield the greatest energy impacts. Actual energy impacts are expected to be much lower. The energy impact of standards based on wet controls, therefore, is minimal. Standards based on dry controls, however, would have no energy impact.

Although wet controls do result in a small adverse impact on gas turbine efficiency, the costs associated with this increased fuel consumption may, for some applications, be partially offset by an increase in the gas turbine's rated power output capability. Based on manufacturer's estimates, gas turbine baseload capacity will be increased by 3 to 4 percent as a result of water injection. In applications where turbines are operated at maximum capacity, such as utility power generation and pipeline compressor stations, this increased baseload capacity essentially reduces the installed costs per kilowatt by the percentage increase in the capacity of the unit, thus slightly reducing the cost impact of standards based on wet controls.

The economic impacts associated with standards based on either wet or dry controls would be small. Dry control costs are difficult to quantify. Many manufacturers, however, have indicated that the cost of dry controls would not exceed the cost of wet controls. Consequently, the analysis of the economic impact of standards of performance is based on the costs of wet controls and assumes that the costs of dry controls, and hence the economic impact of standards based on dry controls, would be comparable. Standards of performance, therefore, based on either wet or dry controls would increase the capital cost of a gas turbine for most applications by about 1 to 4 percent. For offshore industrial applications where desalinization equipment is required to provide water for wet controls, standards would result in a 7 percent increase in the capital cost of a gas turbine. Annualized costs for all applications would be increased from 1 to 4 percent, with utility applications realizing less than a 2 percent increase.

Although it is unlikely that a stationary gas turbine would, of necessity, be installed in an arid area, an analysis was performed which assumed that water would have to be transported to the gas turbine site by truck over a distance of 50 miles. This unlikely situation would result in less than a 4 percent increase in the annualized cost of the gas turbine.

Standards of performance based on wet controls would increase the total capital investment requirements for all industrial and commercial users of large stationary gas turbines (greater than 10,000 hp) by about 36 million dollars by 1982. Total annualized costs would be increased by about 11 million dollars per year in 1982. Standards of performance based on wet controls would have an additional economic impact on users of small stationary gas turbines (less than 10,000 hp) beginning in 1982. Thus, for the period of 1982 through 1987, the capital investment requirements for all stationary gas turbine users would be about 67 million dollars. The annualized costs would be about 30 million dollars by 1987. These impacts would translate into price increases for the end products or services provided by these industrial and commercial users of stationary gas turbines ranging from less than 0.01 percent in the petroleum refining industry to about 0.1 percent in the electric utility industry. Thus, the economic impact of standards of performance based on wet controls would be very small.

Standards of performance based on dry controls would have no economic impact on users by 1982. Following 1982, however, the economic impact of standards based on dry controls would be comparable to that of standards based on wet controls.

Based on this assessment of the impacts of standards of performance based on wet controls and dry controls, wet controls are selected as the "...best system of emission reduction (considering cost)..." for the reduction of NO_x.

As discussed in chapter 4, there are two possible control techniques for reducing SO_2 emissions from stationary gas turbines: flue gas desulfurization (FGD) and the firing of low sulfur fuels. FGD, however, as pointed out in chapter 7, would cost about two to three times as much as the gas turbine. The economic impact of standards of performance based on FGD, therefore, is not considered reasonable.

Low sulfur fuels, such as premium distillate oils or natural gas, are now being burned by nearly all stationary gas turbines. These premium fuels are being burned primarily because the increased maintenance costs associated with firing heavy fuel oils are greater than the savings that would be realized by buying these less expensive heavy or residual fuel oils. Over the next five to ten years, however, as oil prices continue to escalate, the price differential between premium distillate fuel oils and heavy fuel oils will increase. The economic incentive to burn the premium fuels, therefore, will probably become marginal.

In the absence of regulations requiring gas turbines to fire specific fuels, the choice between firing either premium distillate fuel oils or heavy fuel oils will likely be decided on the basis of the relative convenience and availability of these fuels. Premium distillate fuel oils are more convenient to burn than heavy fuel oils because they have a lower viscosity and are easier to handle. Heavy fuel oils frequently require heating, for example, to reduce their viscosity to the point where they can be readily pumped from one location to another. Even if the price differential between premium distillate fuel oils and heavy fuel oils were to increase to the point where the firing of heavy fuel oils was marginally attractive, the greater inconvenience of scheduling and performing the additional maintenance would probably cause a gas turbine user to choose to fire the premium distillate fuel oil. On the basis of convenience, therefore, stationary gas turbines are likely to continue firing premium distillate fuel oils even if the economic incentive to do so becomes marginal.

The impact on ambient air quality of standards of performance based on the firing of low sulfur premium distillate fuel oils in gas turbines, therefore, would be negligible. The economic impact would also be negligible for the same reason and there would be no water, energy, solid waste or noise impact associated with standards based on the firing of low sulfur premium distillate fuel oils.

Based on this assessment of the impacts of standards of performance based on the firing of low sulfur fuel oils, this control technique is selected as "...the best system of emission reduction (considering costs)..." for the reduction of SO_2 emissions.

8.5 SELECTION OF FORMAT FOR THE STANDARDS

A number of different formats could be selected to limit NO_x emissions from stationary gas turbines. Mass standards limiting emissions in terms of power output (i.e. mass of emissions per unit of power output) or concentration standards limiting the concentration of emissions in the exhaust gases discharged into the atmosphere could be developed.

While mass standards may appear more meaningful in the sense that they relate directly to the quantity of emissions discharged into the atmosphere, enforcement of mass standards is more costly and the results more subject to error than enforcement of concentration standards. Determining mass emissions, for example, requires measurement of power output and exhaust gas flow rates in addition to the measurements required for a concentration standard. Power output can be readily obtained at most electric utility installations, but shaft power at compressor and industrial installations would be difficult and expensive to measure accurately. Also, the high turbulence characteristic of gas turbine exhaust gases makes the determination of exhaust gas flow rates, and hence mass emissions, subject to considerable error. Manipulation of this data increases the number of calculations necessary, compounding the errors inherent within the data and increasing the chance for human error.

Enforcement of concentration standards, however, requires a minimum of data and calculations, decreasing the costs and minimizing the chances for error in determining compliance.

The primary disadvantage normally associated with concentration standards is that of possible circumvention by dilution of the exhaust gases discharged to the atmosphere lowering the concentration of emissions, but not reducing the total mass emitted. Thus, concentration standards must be written to insure that the standards are not met merely by addition of dilution air. For combustion processes, this can be accomplished by correcting measured concentrations to a reference concentration of O_2. The O_2 concentration in the exhaust gases is related to the excess (or dilution) air. Typical O_2 concentrations in gas turbine exhaust gases are about 15 percent. Thus, referencing the standard to 15 percent oxygen effectively precludes circumvention by dilution. Consequently, a concentration standard referenced to 15 percent oxgen is selected as the format for standards of performance for stationary gas turbines.

Selection of a concentration format, however, could penalize high efficiency gas turbines. Higher efficiencies are normally achieved by increasing combustor operating pressures and temperatures and NO_x formation generally increases exponentially with increased pressure and temperature. High efficiency turbines, therefore, generally discharge gases with higher NO_x concentrations than low efficiency turbines. A concentration standard based on low efficiency turbines could restrict the use of some high efficiency turbines. Conversely, a concentration standard based on high efficiency turbines could allow such high NO_x concentrations that low efficiency turbines would require no controls. Consequently, having selected a concentration format for standards of performance, an efficiency adjustment factor needs to be selected to permit higher NO_x emissions from high efficiency gas turbines.

As mentioned above, NO_x emissions tend to increase exponentially with increased efficiency. It is not reasonable from an emission control viewpoint, however, to select an exponential efficiency adjustment factor. Such an adjustment would at some point allow very large increases in emissions for very small increases in efficiency. The objective of an efficiency adjustment factor should be to give an emissions credit for the lower fuel consumption of high efficiency gas turbines. Since the relative fuel consumption of gas turbines varies linearly with efficiency, a linear efficiency adjustment factor is selected to permit increased NO_x emissions from high efficiency gas turbines. A linear efficiency adjustment factor also effectively limits NO_x emissions to a constant mass emission rate per unit of power output.

The efficiency adjustment factor needs to be referenced to a baseline efficiency. Since most existing simple cycle gas turbines fall in the range of 20 to 30 percent efficiency, 25 percent is selected as the baseline efficiency. The efficiency of stationary gas turbines is usually expressed in terms of heat rate which is the ratio of heat input, based on lower heating value (LHV) of the fuel, to the mechanical power output. The heat rate of a gas turbine operating at 25 percent efficiency is 14.4 kilojoules per watt·hr (10,180 Btu per hp·hr). Thus, the following linear adjustment factor is selected to permit increased NO_x emissions from high efficiency stationary gas turbines:

$$x_a = x \frac{14.4}{Y}$$

where:

x_a = adjusted NO_x emissions permitted at 15 percent oxygen and ISO conditions, ppmv.

x = NO_x emission limit specified in the standards at 15 percent oxygen and ISO conditions, (i.e. 75 ppmv).

Y = LHV heat input per unit of power output (kilojoules/watt·hr).

NOTE: ISO conditions refers to standard atmospheric conditions of 760 mm mercury, 288° Kelvin and 60 percent relative humidity.

The only intent of this efficiency adjustment factor is to permit a linear increase in NO_x emissions with increased efficiencies above 25 percent. Consequently, the adjustment factor would not be used to adjust the emission limit downward for gas turbines with efficiencies of less than 25 percent.

The selection of this efficiency adjustment factor will essentially preclude the development of gas turbines which achieve higher operating efficiencies at the expense of exponential increases in NO_x emissions. As a result, this linear adjustment factor will require the development of effective NO_x controls on future high efficiency gas turbines.

The rationale for the selection of a format for the limiting of SO_x emissions from gas turbines is much the same as that discussed above for NO_x emissions. Thus, to be consistent with the format selected for standards limiting NO_x emissions, a concentration standard is chosen as the format for the SO_2 standard. An emission limit in terms of percent fuel sulfur content has also been included in the SO_2 standard to give the owner or operator the flexibility of either measuring the SO_2 concentration of the exhaust gas or analyzing the fuel being fired in the turbine. Either format for the SO_2 standard can be used since nearly all of the sulfur in the fuel is converted to SO_2.

The efficiency factor associated with the NO_x emission limit, however, will not apply to the SO_2 emission limit because SO_2 emissions do not vary with turbine efficiency.

8.6 SELECTION OF EMISSION LIMIT

Selection of the NO_x emission limit is based on the data and information discussed in chapter 4. A detailed tabulation of the data may be found in appendix C, Figure 4-13 (page 4-25), which is a summary of the wet control data reproduced here as Figure 8-1.

While all of these data do not represent maximum NO_x reduction efforts, the data do indicate the general range of controlled NO_x emissions. Considering only the data which represent major NO_x control efforts (i.e.,

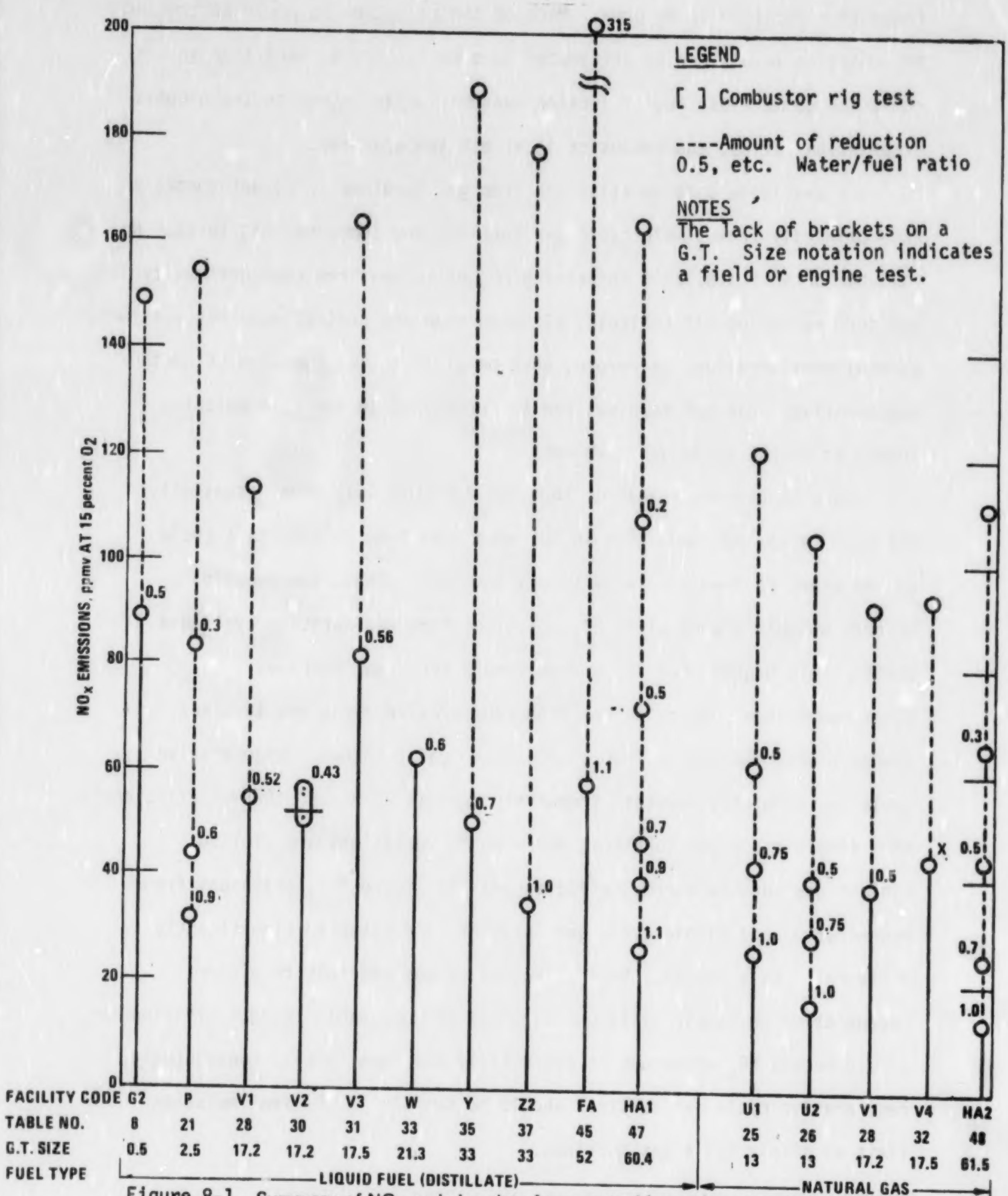

Figure 8-1. Summary of NO_x emission data from gas turbines using wet control techniques.

water-to-fuel ratios of about 0.6 or greater), the controlled NO_x emissions range from about 15 to 60 ppmv. Much of the variation in these controlled NO_x emission levels can be attributed to such factors as variation in combustor geometries, fuel injection systems, water injection techniques, compression ratios, and combustor inlet air temperatures.

The available data on emissions from gas turbines using wet controls come primarily from simple cycle gas turbines and combustor rig tests. No reliable data is available concerning NO_x emissions from regenerative cycle gas turbines using wet controls, although some dry control data was obtained. Careful consideration, therefore, must be given to the question of whether regenerative cycle gas turbines can be controlled to the same emission levels as simple cycle gas turbines.

There is general agreement that wet controls will give essentially the same percentage reduction in NO_x emissions from regenerative cycle gas turbines as from simple cycle gas turbines. Thus, the question becomes whether uncontrolled NO_x emissions from regenerative cycle gas turbines are higher than those from simple cycle gas turbines. On first comparison, NO_x emissions from regenerative cycle gas turbines appear higher than those from simple cycle gas turbines. Regenerative cycle gas turbines, however, frequently operate at higher thermal efficiencies than simple cycle gas turbines, and when NO_x emissions are plotted against gas turbine thermal efficiency as in Figure 8-2, emissions from regenerative and simple cycle gas turbines do not appear significantly different. As a result, the application of wet controls to either regenerative or simple cycle gas turbines of comparable thermal efficiencies should reduce NO_x emissions to essentially the same level. Consequently, regenerative cycle gas turbines should be subject to the same emission limit as simple cycle gas turbines.

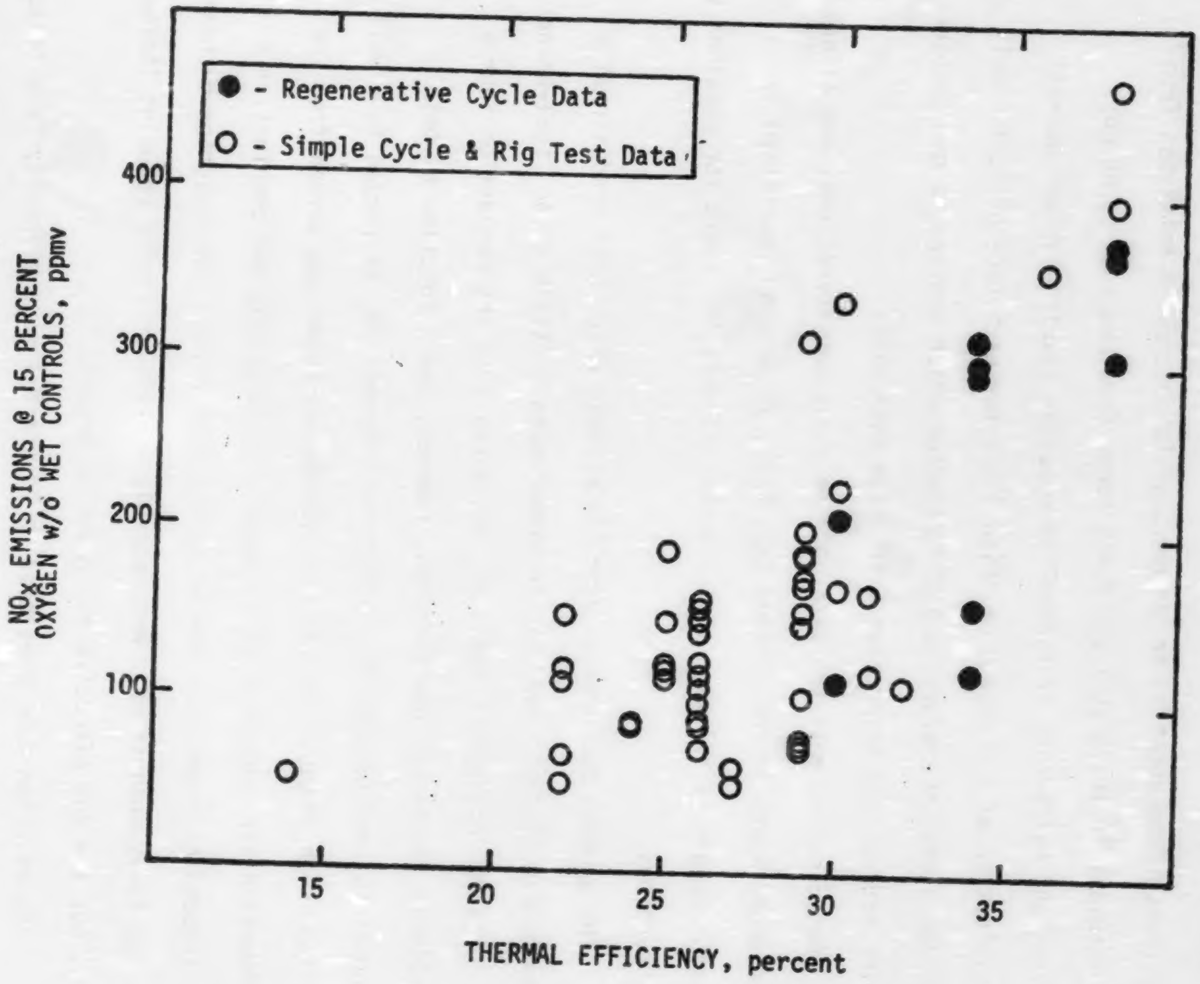

Figure 8-2. Relationship between efficiency and uncontrolled NO_x emissions.

The data also indicate that gas turbines firing gaseous fuels typically have slightly lower controlled NO_x emission levels than gas turbines firing distillate fuels. Again, considering only the data representing major NO_x control efforts, controlled emissions from gas turbines firing gaseous fuels range from about 15 to 50 ppmv, while controlled emissions from gas turbines firing distillate fuels range from about 25 to 60 ppmv. This slight difference in controlled emission levels does not warrant the selection of a separate emission limit for each type of fuel. Only one emission limit, therefore, will be selected which applies to gas turbines firing natural gas or premium distillate fuel oils.

Based on the emission data and the above considerations, and allowing for some uncertainty in the limited data base, 75 ppmv NO_x corrected to 15 percent oxygen is selected as the numerical emission limit for stationary gas turbines.

The gaseous and premium distillate fuels which have traditionally been burned in stationary gas turbines contain little or no "fuel-bound" or "organic" nitrogen. Total NO_x emissions from any combustion source including stationary gas turbines, however, are a function of both thermal NO_x and organic NO_x formation. Thermal NO_x is formed in a well defined high temperature reaction between nitrogen and oxygen from the combustion air. Organic NO_x, however, is formed by the combination of fuel-bound nitrogen with oxygen during combustion. The reaction mechanism is not fully understood. Wet controls are effective for reducing thermal NO_x, but are not effective for reducing organic NO_x.

The emission data presented in Figure 8-1 come primarily from tests on stationary gas turbines firing traditional premium gaseous or distillate fuels which contain little or no fuel-bound nitrogen. The measured NO_x

emissions, therefore, are comprised primarily of thermal NO_x. Thus, the numerical emission limit selected above is not appropriate for the firing of fuels containing significant amounts of fuel-bound nitrogen.

Figure 8-3 illustrates the variation in the fuel-bound nitrogen content of various petroleum fuels. Although this figure is based on limited data, it shows the large difference between the fuel-bound nitrogen levels of the premium distillate fuels and various heavy petroleum fuels. Generally speaking, heavy petroleum fuels are not readily available with low fuel-bound nitrogen levels. Crude oils generally range between 0.1 and 0.2 percent nitrogen. As a point of reference, this figure indicates that about half of all heavy petroleum fuels contain less than 0.25 percent nitrogen.

Quantifying the organic NO_x contribution to total gas turbine NO_x emissions is complicated by the fact that the percentage of fuel-bound nitrogen converted to organic NO_x varies with the fuel-bound nitrogen level. Figure 3-37 (page 3-88), reproduced here as Figure 8-4, illustrates the variation in conversion of fuel-bound nitrogen to organic NO_x with fuel-bound nitrogen level of the fuel. While this figure is also based on very limited data, it indicates that the percentage of fuel-bound nitrogen converted to organic NO_x decreases as fuel-bound nitrogen level increases. Below a fuel-bound nitrogen level of about 0.05 percent, essentially 100 percent is converted to NO_x. Above a fuel-bound nitrogen level of about 0.4 percent, only about 40 percent is converted to NO_x. Using Figure 8-4, an estimate of the effect on controlled NO_x emission levels of firing fuels with various fuel-bound nitrogen levels in gas turbines is presented in Figure 8-5. This figure illustrates both the organic NO_x and thermal NO_x contributions to total NO_x emissions, assuming that thermal NO_x is controlled to 75 ppmv through the use of wet controls. As fuel-bound nitrogen level increases, the organic NO_x contribution becomes increasingly

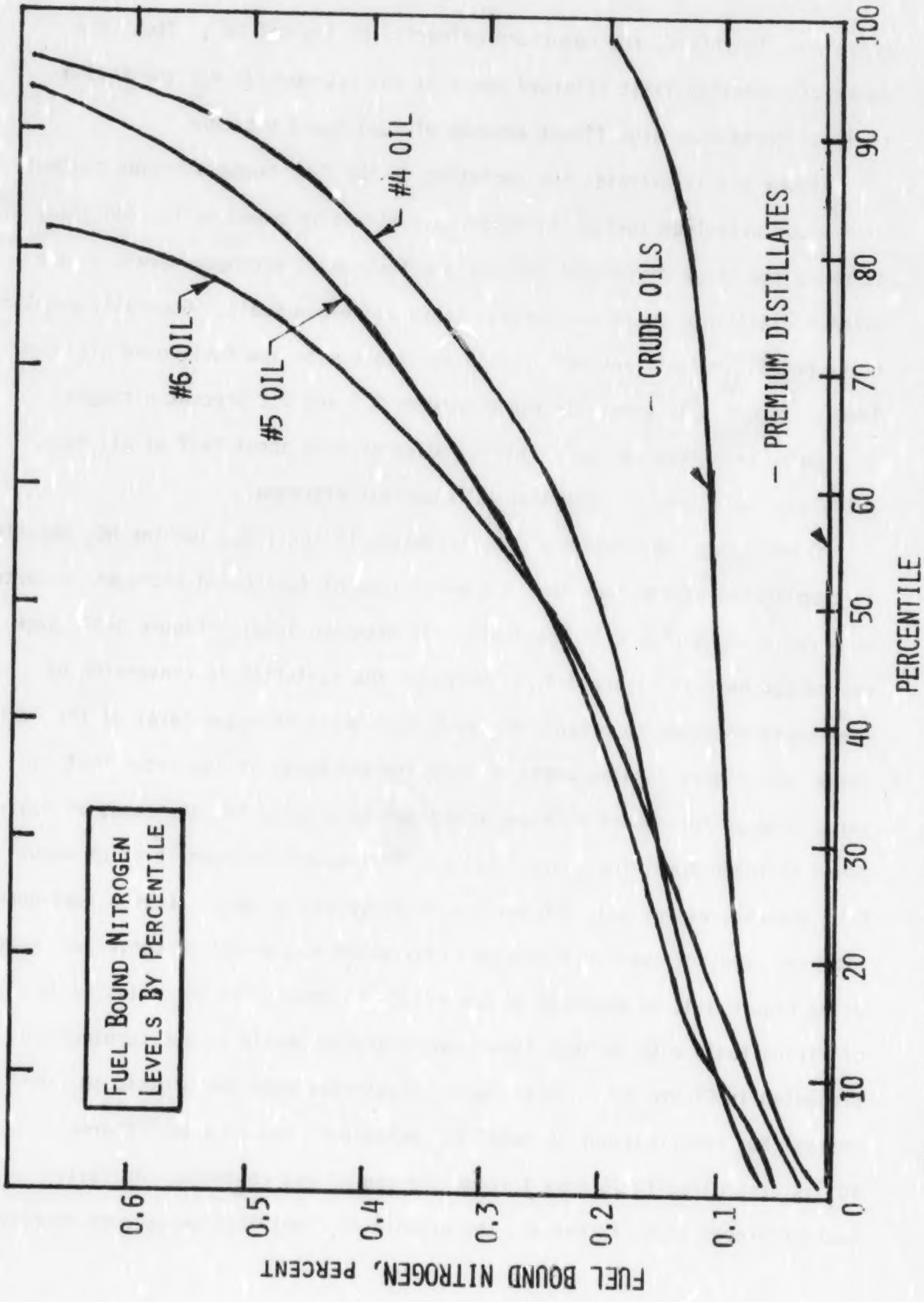

Figure 8-3. Variation in fuel-bound nitrogen content of petroleum fuels.

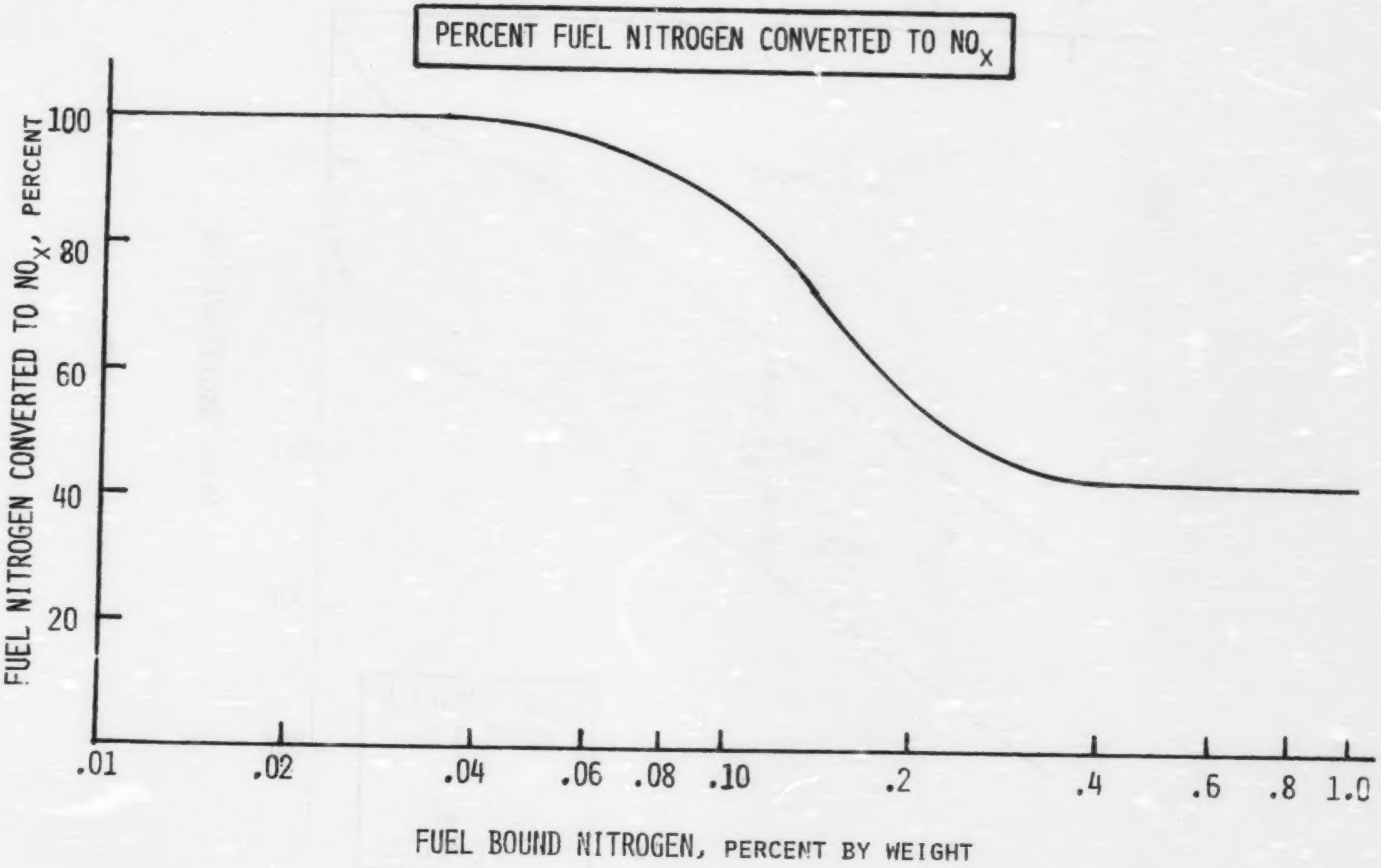

Figure 8-4. Variation in conversion of fuel-bound nitrogen to organic NO_x

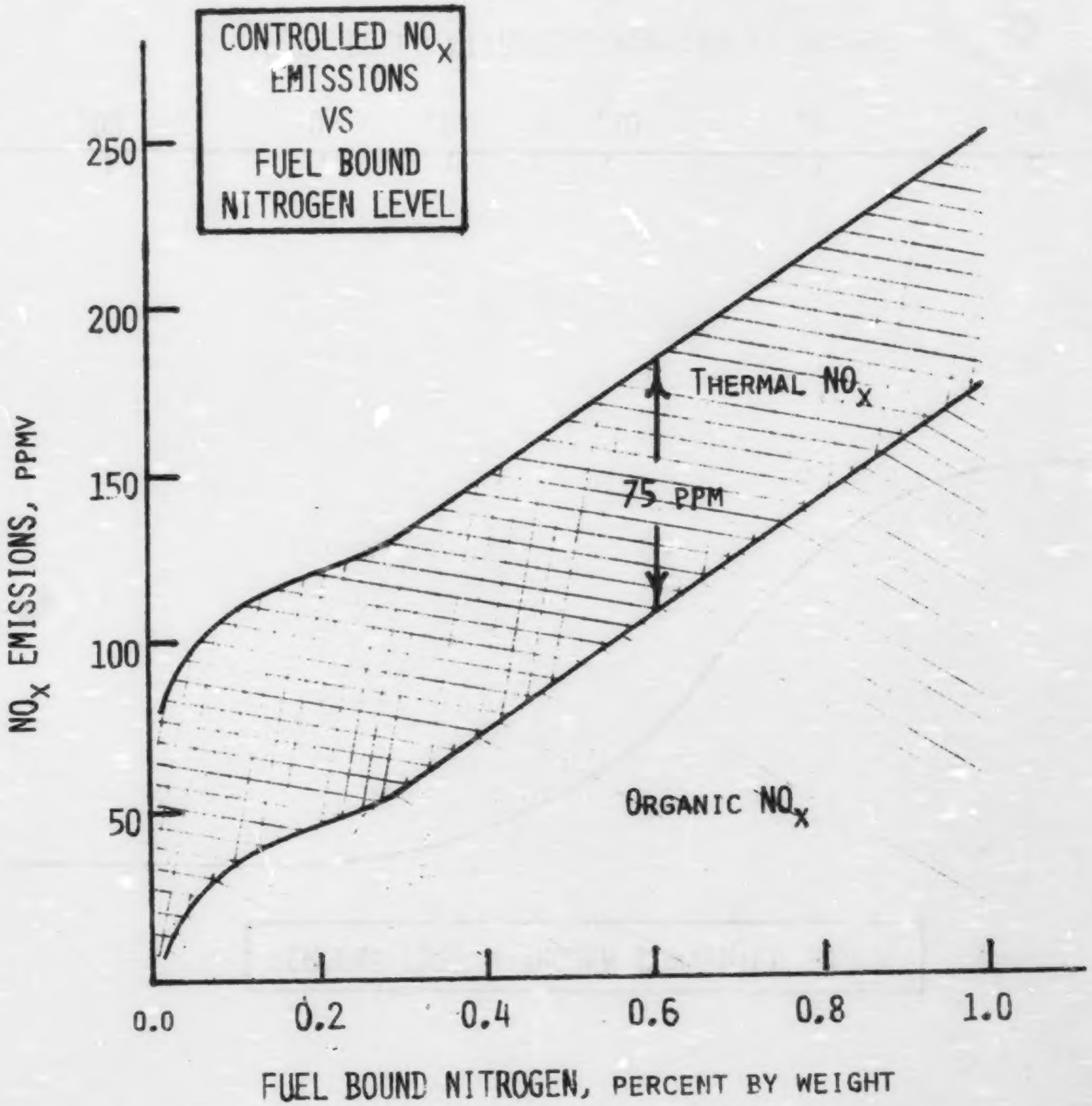

Figure 8-5. Controlled NO_x emission levels vs fuel-bound nitrogen level

significant to the point where organic NO_x emissions are greater than controlled thermal NO_x emissions.

Three alternative approaches emerge to address the fuel-bound nitrogen contribution to total NO_x emissions from gas turbines. The first alternative is to exempt heavy fuels from standards of performance. This approach would allow gas turbines firing heavy fuels to operate with no emission controls. In addition to the difficulties of distinguishing between premium and heavy fuels in the standards, this approach would tend to encourage users to burn heavy fuels as means of evading standards of performance.

The second alternative is to base standards of performance on the firing of low nitrogen fuels. This approach would require emission controls on all new stationary gas turbines, but would effectively preclude the firing of fuels other than those premium gaseous and distillate fuels which turbines are now using. Firing of heavy fuels would require major breakthroughs in controlling the contribution of fuel-bound nitrogen to NO_x formation.

The third alternative is to include an allowance in the NO_x emission limit which is a function of the fuel-bound nitrogen level of the fuel fired. This approach would require NO_x controls on all new stationary gas turbines, but would not restrict new gas turbines to firing premium gaseous and distillate fuels. Thus, new stationary gas turbines would not be penalized for firing heavy fuels, nor would there be any added impetus toward the firing of heavy fuels in order to evade standards of performance.

As discussed earlier, low sulfur fuels, such as premium distillate fuel oils or natural gas, are now being fired by nearly all stationary gas turbines. These premium fuels are being burned primarily because the increased maintenance costs associated with firing heavy fuel oils are greater than the savings that would be realized by buying these less expensive heavy or residual fuel oils. Over the next five to ten years, however, as oil prices continue to escalate, the price differential between premium distillate fuel oils and heavy fuel oils will probably increase and the economic incentive to burn the premium fuel oils will probably become marginal. It is also possible, however, that there could be limited supplies of premium distillate fuel oils due to declining production of oil and natural gas in the United States, increased demands for these premium fuels by users other than gas turbines which cannot utilize heavy or residual fuel oils, and the uncertainty of additional crude oil supplies in the world energy markets. In the event of limited supplies, many new gas turbines would probably be designed to fire residual or heavy fuel oils. Consequently, in order to provide gas turbine owners and operators the flexibility to fire either premium or heavy and residual fuel oils, but to ensure that standards of performance add no impetus toward the firing of heavy fuel oils as a means of evading standards, alternative three is selected for standards of performance limiting NO_x emissions from stationary gas turbines.

An allowance in the NO_x emission limit dependent on fuel-bound nitrogen level with no upper limit on emissions, however, could permit extremely high NO_x emissions when firing some very high nitrogen-containing fuels. Thus, it is essential that restraints be placed on such an emission allowance. Using the data presented in Figures 8.3, 8.4 and 8.5, a fuel-bound nitrogen allowance can be developed that allows approximately 50 percent availability of the heavy fuel oils. This corresponds to a fuel-bound nitrogen content of about 0.25 weight percent. Referring to Figure 8.5, firing a fuel with 0.25 weight percent fuel-bound nitrogen increases controlled NO_x emissions by about 50 ppm. Consequently, a fuel-bound nitrogen NO_x emission allowance based on a straight line approximation of Figure 8.5, with a maximum allowance of 50 ppm, is selected for standards of performance.

The numerical NO_x emission limit, therefore, is a function of thermal efficiency of the gas turbine and fuel-bound nitrogen level of the fuel fired as follows:

$$NO_x = [0.0075\ (E) + F]$$

where:

NO_x = allowable NO_x emissions (percent by volume at 15 percent oxygen).

E = the efficiency adjustment factor:

$$\frac{14.4\ \text{kilojoules/watt}\cdot\text{hr}}{\text{LHV heat input per unit of power output}}$$

F = the fuel-bound nitrogen allowance:

Fuel-Bound Nitrogen (percent by weight)	F
$N \leq 0.015$	0
$0.015 < N \leq 0.1$	0.04 (N)
$0.1 < N \leq 0.25$	0.004 + 0.0067 (N-0.1)
$N > 0.25$	0.005

The effect of ambient atmospheric conditions on NO_x emissions from stationary gas turbines is substantial. Large changes in relative humidity, for example, can cause NO_x emissions to vary by a factor of 2 or more. In order to insure that standards of performance are enforced uniformly, therefore, the effect of ambient atmospheric conditions on NO_x emission levels needs to be taken into account. The following equation to correct measured NO_x emissions to ISO ambient atmospheric conditions was derived by extracting the common elements from several ambient correction factors proposed by gas turbine manufacturers. This correction factor, therefore, represents the general effect of ambient atmospheric conditions on NO_x emissions. Consequently, this correction factor, or an alternative factor as discussed below, will be used to adjust measured NO_x emissions during any performance test to determine compliance with the numerical emission limit.

$$NO_x = (NO_{x_{obs}})\left(\frac{P_{ref}}{P_{obs}}\right)^{0.5} e^{19(H_{obs} - 0.00633)}$$

where:

NO_x = emissions of NO_x at 15 percent oxygen and ISO standard ambient conditions.

$NO_{x_{obs}}$ = measured NO_x emissions at 15 percent oxygen, ppmv.

P_{ref} = reference combustor inlet absolute pressure at 101.3 kilopascals (1 atmosphere) ambient pressure.

P_{obs} = measured combustor inlet absolute pressure.

H_{obs} = specific humidity of ambient air.

e = transcendental constant (2.718).

As an alternative, gas turbine manufacturers may elect to develop custom correction factors for adjusting measured NO_x emissions from particular gas turbine models to ISO standard ambient conditions of pressure (1 atmosphere), humidity (60 percent relative humidity), and temperature (288°K). Some gas turbine manufacturers have proposed ambient correction factors which include variables such as fuel-to-air ratios and combustor temperatures. These variables are difficult to measure and are operating parameters which may vary widely due to factors other than ambient conditions. For this reason, any custom correction factor must be developed in terms of only the following variables: combustor inlet pressure, ambient air pressure, ambient air humidity, and ambient air temperature. All such correction factors must be substantiated with data and then approved by EPA for use in determining compliance with the NO_x emission limit. The ambient correction factor will be applied to all performance tests, not only those in which the use of such factors would reduce measured emission levels.

As discussed in section 8.4, some delay is required before this emission limit can be applied to small stationary gas turbines. This delay in effective date is to provide time for manufacturers to incorporate NO_x controls on their small production stationary gas turbine models. It is estimated that about three years delay in the effective date of the standard for small stationary gas turbines would be required to allow manufacturers time to incorporate and test wet controls on these gas turbines. Some manufacturers have expressed optimism at being able to meet this emission limit using dry controls if given about five years delay. Since these small turbines represent only about 10 to 15 percent of the total NO_x emissions from stationary gas turbines, the difference in environmental impact of a three-year versus five-year delay would be quite small. Additionally, a three-year delay would essentially force these manufacturers to incorporate wet controls whereas a five-year delay would provide the flexibility to use wet controls or to develop and use dry controls. Consequently, five years is selected as the delay period for implementation of these standards on small stationary gas turbines.

In selecting the size cutoff to differentiate between large and small stationary gas turbines, consideration must be given to the purpose for the cutoff and the effect on competitive markets. The purpose of the cutoff is to differentiate between large gas turbines where wet controls have been commercially demonstrated and small gas turbines where wet controls, although effective, have not been generally applied on a commercial basis. Consideration of the market data in chapter 3 reveals that there are two major competitive markets for stationary gas turbines which can be generally described as small gas turbines and large gas turbines. The size range of 5000 to 10,000 horsepower essentially separates these two markets. All gas turbines above this range are manufactured by companies which have developed wet control sytems for their stationary gas turbines. The size cutoff, therefore, between small and large gas turbines is selected as the upper end of this range. Thus, large stationary gas turbines are defined as those with heat input greater than 107.2 gigajoules per hour (approximately 10,000 horsepower for a 25 percent efficient gas turbine).

As discussed earlier in this chapter, SO_2 emissions from a stationary gas turbine depend on the sulfur content of the fuel fired. The best system of emission reduction, considering costs, selected for SO_2 emissions was the firing of low sulfur fuels. Selection of the SO_2 emission limit, therefore, will be based on the use of this control system.

As also discussed earlier, nearly all stationary gas turbines are currently firing natural gas or premium distillate fuel oil; although over the next five to ten years, some new gas turbines may fire heavy or residual fuel oil for either economic reasons or if a shortage in supply of premium fuel oils should develop. A fuel-bound nitrogen allowance to permit increased NO_x emissions has been selected to allow turbines to burn approximately 50 percent of currently available heavy fuels. To be consistent with the objective of the fuel-bound nitrogen allowance, the SO_2 emission limit is selected as 150 ppm referenced to 15 percent O_2. This corresponds to a fuel sulfur content of approximately 0.8 percent by weight and would allow about 50 percent availability of heavy fuel oils.

The five-year delay of the NO_x emission limit applied to small gas turbines (less than 10,000) to provide manufacturers time to incorporate wet controls onto their turbines does not apply to the SO_2 emission limit since the control technique of burning low sulfur fuels is available to all turbines at the present time.

8.7 MODIFICATION/RECONSTRUCTION

A discussion of the modification and reconstruction regulations and how they pertain to the gas turbine industry can be found in chapter 5. Since few modified or reconstructed gas turbines are anticipated, the modification and reconstruction regulations will have little impact. Wet controls, however, are as effective in reducing emissions of NO_x from modified or reconstructed gas turbines as from new gas turbines. Thus, modified or reconstructed gas turbines merit no special allowance in the standards of performance.

8.8 SELECTION OF MONITORING REQUIREMENTS

To provide a convenient means for enforcement personnel to ensure that an emission control system installed to comply with standards of performance is properly operated and maintained, monitoring requirements are generally included in standards of performance. For stationary gas turbines the most straightforward means of ensuring proper operation and maintenance is to monitor emissions released to the atmosphere.

EPA has established NO_x monitoring performance specifications in Appendix B of 40 CFR Part 60 for large industrial sources with well developed velocity and temperature profiles. Stationary gas turbines, however, do not have well developed velocity and temperature profiles in all cases. Gas stratification of the turbine exhaust, for example, makes the location of the sample point critical. Also, since some turbines are started remotely from a central location, special systems and data reporting procedures would be necessary to start and maintain continuous monitors.

Currently there are no NO_x continuous monitors operating on gas turbines, and resolution of these sampling problems and development of performance specifications for continuous monitoring systems would entail a major development program. For these reasons, continuous monitoring of NO_x emissions from gas turbines will not be required by the new source performance standard.

A means of ensuring operation of the water injection system used to control NO_x emissions from gas turbines is to monitor the water-to-fuel ratio being fed to the turbine. Both water and fuel monitors are readily available and are demonstrated technology for use on gas turbines. Consequently, to ensure operation of water injection systems, the standards for stationary gas turbines will require continuous monitoring of the water-to-fuel ratio where water injection is employed to comply with the NO_x standard.

Also, a means of ensuring the firing of fuels with the proper nitrogen content to control NO_x emissions caused by fuel-bound nitrogen is to monitor the nitrogen content of the fuel being fired. Consequently, any owner or operator that uses the fuel-bound nitrogen allowance to comply with the NO_x emission limit will be required by the standard to monitor the nitrogen content of the fuel.

The continuous monitoring of SO_2 emissions will not be required by the new source performance standard for the same reasons continuous monitoring of NO_x emissions will not be required. A means of ensuring the firing of low sulfur fuels to control SO_2 emissions, however, is to monitor the sulfur content of the fuel being burned. This is already a common practice among gas turbine users. Consequently, to ensure the use of low sulfur fuels by stationary gas turbines to comply with the SO_2 emission limit, the standard will require monitoring of the sulfur content of the fuel.

8.9 SELECTION OF PERFORMANCE TEST METHODS

Reference Method 20, "Determination of Nitrogen Oxides, Sulfur Dioxide, and Oxygen Emissions from Stationary Gas Turbines," is selected as the performance test method to determine compliance with standards of performance limiting NO_x emissions for stationary gas turbines. This test method is based on the EPA gas turbine field tests and on background data for continuous monitoring system specifications (Federal Register, October 6, 1975). Reference Method 20 includes (1) measurement system design criteria, (2) measurement system performance specifications and performance test procedures, and (3) procedures for emission sampling. The performance specifications include the span drift, zero drift, linearity check, response time of the system, and interference checks. This method allows a choice of instruments and will provide reliable data if the performance specifications are met. Appendix D, section 3 and Appendix G give a full explanation of Reference Method 20.

The Mobile Source and SAE test procedures were considered as possible performance test methods but were rejected because they specified the use of particular types of instruments, rather than design criteria and performance specifications of the measurement system. Both the SAE and Mobile Source test methods, however, are acceptable alternative methods, if the selected instrument models are capable of meeting the performance specifications of Reference Method 20.

As mentioned earlier in this chapter, the NO_x emissions measured by Reference Method 20 will be affected by ambient atmospheric conditions. Consequently, measured NO_x emissions will be adjusted during any performance test to determine compliance by the following equation or by custom equations developed by turbine manufacturers, owners or operators and approved by the Administrator.

$$NO_x = (NO_{x_{obs}}) \left(\frac{P_{ref}}{P_{obs}}\right)^{0.5} e^{19 (H_{obs} - 0.0633)}$$

where:

NO_x = emissions of NO_x at 15 percent oxygen and ISO standard ambient conditions.

$NO_{x_{obs}}$ = measured NO_x emissions at 15 percent oxygen, ppmv.

P_{ref} = reference combustor inlet absolute pressure at 101.3 kilopascals (1 atmosphere) ambient pressure.

P_{obs} = measured combustor inlet absolute pressure.

H_{obs} = specific humidity of ambient air.

e = transcendental constant (2.718).

In order to apply the fuel-bound nitrogen allowance included as part of the NO_x emission limit, the nitrogen content of the fuel must be determined. The analytical methods and procedures employed to determine the nitrogen content of the fuel shall be approved by the Administrator and shall be accurate to within plus or minus five percent.

In lieu of determining the SO_2 concentration of the emission gas from the turbine by using Method 6, compliance with the standard may be determined by determining the sulfur content of the fuels being used by the gas turbine. Sulfur content of the fuel will be determined using ASTM D2880-71 for liquid fuels and ASTM D1072-70 for gaseous fuels.

References

1. Responses from 17 electric utilities submitted by Baruch, S.B., Edison Electric Institute, to K.R. Durkee, EPA. October 1975-January 1976.

APPENDIX A. EVOLUTION OF THE SELECTION OF THE BEST SYSTEM OF EMISSION REDUCTION

INTRODUCTION

The study to develop proposed standards of performance for new stationary gas turbines began in 1971. In the course of the program, a literature search was conducted and contacts were made with practically all domestic manufacturers of stationary gas turbines, several foreign manufacturers, a number of users (electrical utilities and pipeline companies), technical societies, trade associations, two trade journals and several air pollution control districts.

EPA personnel met with many of these interested parties on numerous occasions and exchanged much correspondence and countless telephone calls. The dates and locations of the meetings are documented in Table A-1, but the letters and telephone calls were too numerous to catalog.

Considerable data was obtained from published reports and trade publications. Direct contacts with engine manufacturers also provided much of the data on emissions from uncontrolled engines, on control technologies, and on emissions from controlled engines. Additional information regarding stationary applications, control technologies, and their costs were received from manufacturers in response to official requests for data by the Director, Emission Standards and Engineering Division, U.S. EPA. These requests were sent to manufacturers under the authority specified in Section 114 of the Clean Air Act. Most of the data on the effectiveness of the various dry controls for NO_x came from laboratory experiments at the manufacturer's plants. Much of the test data on wet controls for NO_x, however, is from field tests by manufacturers and users. For many of the field tests the control equipment was a temporary installation and was utilized only for the duration of the tests. EPA also conducted tests of gas turbines at two electrical utilities to validate the test methods and develop corroborating data.

The direct contacts were supplemented by visits to the manufacturers listed on Table A-2. The purpose of these visits was to obtain more information than possible over the telephone or by letter concerning the status of their R&D efforts in emission reductions, their experience with the commonly proposed control technologies, their estimates of the cost and time required to incorporate such controls on their engines, and the importance of the stationary market to them.

Standards Support Documents were prepared and standards with slight variations were recommended by the Industrial Studies Branch (ISB), ESED, to the National Air Pollution Control Techniques Advisory Committee (NAPCTAC) on three occasions:

February 21, 1973
May 30, 1973
January 9, 1974

The primary pollutants of concern at that time were SO_2, NO_x, CO and visible emissions.

After the January 9, 1974, meeting with the NAPCTAC, several meetings were held with the industry. New issues were brought up and considerable time was spent in obtaining technical and economic data to resolve them. In August 1974, a completely new Standards Support Document was completed. In October 1974, final preparation of the wording of the regulation and its preamble was begun. After some initial work, the project was delayed several months because of higher priority work. During this time period, EPA expanded the requirements for a Standards Support Document to include inclusion of an abbreviated environmental impact statement. This required the entire document (now called a Standards Support and Environmental Impact Statement) to be rewritten, a project which was begun in September 1975.

Since more than a year had passed since the last Standards Support Document was prepared (August 1974), several manufacturers and users of gas turbines were contacted and recent literature was reviewed to determine if any major changes had occurred either in control techniques or the market. There were several, such as:

A. Fuels were no longer scarce because the oil embargo had ended.

B. The trend to burn crude and residual oils in gas turbines had been reversed.

C. Several manufacturers had aggressively pursued the development of dry controls (via changes in the combustor geometry) to reduce peak flame temperatures and duration and thereby reduce NO_x formation.

D. Many power companies had installed water treatment and injection systems for controlling NO_x emissions from turbines. When the last document was written there were few installations using water injection, over 70 such turbines are now known.

E. Market conditions for gas turbines had changed.

In light of these changes;

1. Eight letters were sent out, under the authority of section 114 of the Clean Air Act, to obtain technical and economic data from gas turbine users and manufacturers.

2. The same request was sent to two pipeline companies, two manufacturers of gas turbines, and two industry associations who volunteered to assist us by supplying information.

3. A contractor was engaged to obtain current information on economic and market data.

The information resulting from these efforts is incorporated into the body of this report and the data are summarized in Appendix C.

Table A-1. MEETING RECORD

	Date	Company or Association	Meeting Location
1	1/23/72	Institute of Gas Technology	Durham, N.C.
2	4/12/72	Westinghouse Electric Corp.	Durham, N.C.
3	4/12/72	Turbo Power and Marine Systems, Pratt & Whitney Aircraft	Durham, N.C.
4	8/17/72	San Diego Air Pollution Control District	San Diego, Cal.
5	10/12/72	Turbodyne Corporation	Durham, N.C.
6	10/27/72	General Electric Company	Durham, N.C.
7	10/31/72	Turbo Power and Marine Systems, Pratt & Whitney Aircraft	Durham, N.C.
8	12/13/72	Turbodyne Corporation	Durham, N.C.
9	12/15/72	Turbo Power and Marine Systems	Durham, N.C.
10	3/26/73	General Motors Corporation	Durham, N.C.
11	4/12/73	Apollo Chemical	Durham, N.C.
12	4/26/73	General Electric Company	Durham, N.C.
13	5/17/73	Southern California Gas Co.	Durham, N.C.
14	6/13/73	Columbia-Willamette Air Pollution Authority	Portland, Ore.
15	6/20/73	General Electric Company	Durham, N.C.
16	8/9/73	Westinghouse Electric Corp.	Durham, N.C.
17	9/6/73	Oregon Dept. of Environmental Quality (DEQ)	Portland, Ore.
18	9/27/73	Bell Labs	Durham, N.C.

	Date	Company or Association	Meeting Location
19	11/16/73	General Electric Company	Durham, N.C.
20	11/28/73	Florida Power & Light Co.	Durham, N.C.
21	3/22/74	American Gas Association	Durham, N.C.
22	5/17/74	Ethyl Corporation	Durham, N.C.
23	6/10/74 - 6/12/74	Air Pollution Control Assoc. Annual Mtg.	Denver, Colo.
24	7/2/74	General Motors, Detroit Diesel Allison Division	Durham, N.C.
25	8/22/74	General Electric Company	Durham, N.C.
26	8/29/74	Solar Division of International Harvester	Durham, N.C.
27	8/30/74	American Nat'l Standards Inst.	Durham, N.C.
28	10/3/74	Airesearch Man. Co. of Arizona	Durham, N.C.
29	10/25/74	Solar Division of International Harvester	Durham, N.C.
30	11/7/74	Westinghouse Electric Corp.	Durham, N.C.
31	1/10/75	General Electric Company	Durham, N.C.
32	1/10/75	Turbo Power & Marine Systems	Durham, N.C.
33	6/10/75	American Gas Association	Durham, N.C.
34	8/19/75	General Electric Company	Durham, N.C.
35	10/14/75	Engine Manufacturers Association	Durham, N.C.
36	1/21/76	Solar Division of International Harvester	Durham, N.C.
37	2/18/76	Edison Electric Institute	Durham, N.C.
38	3/4/76	Turbo Power & Marine Systems	Durham, N.C.
39	3/9/76	General Electric Company	Durham, N.C.
40	8/10/76	NAPCTAC Meeting	Chicago, Ill.

Table A-2. SURVEY & INSPECTION TRIPS

Date	Company	Location	Type of Company
6/6/72	General Electric Company	Schenectady, N. Y.	Mfr
6/7/72	Bell Labs	Murray Hill, N. J.	User
6/8/72	Turbo Power & Marine Systems	Farmington, Conn.	Mfr
6/9/72	Westinghouse Electric Corp.	Philadelphia, Pa.	Mfr
8/16/72	Airesearch Manufacturing Co. of Arizona	Phoenix, Arizona	Mfr
8/17/72	Solar Div. of International Harvester	San Diego, Cal.	Mfr
8/18/72	San Diego Gas & Electric Company	San Diego, Cal.	User (Utility)
2/15/73	Exxon Chemical Co.	Baytown, Texas	User
2/16/73	Exxon Chemical Co.	Baton Rouge, La	User
3/20/73	Naval Air Rework Facility	Jacksonville, Fla.	User
6/11/73	Carolina Power & Light Company	Goldsboro, N. C.	User (Utility)
6/13/73	Portland General Electric	Portland, Oregon	User (Utility)
6/13/73	Portland General Electric	Salem, Oregon	User (Utility)
4/9/74	General Electric Company	Schenectady, N. Y.	Mfr
6/6/74	Detroit Diesel Allison (GM)	Indiannapolis, Ind.	Mfr

Appendix B

Agency Guidelines for Preparing Regulatory Action Environmental Impact Statements (39 FR 37419)	Location Within the Standards Support and Environmental Impact Statement
1. Background and Description of Proposed Action	
Summary of Proposed Standards	The standards are summarized in chapter 1.
Statutory Basis for the Standard	The statutory basis for the standard is given in chapter 2.
Facility Affected	A description of the facility to be affected is given in chapter 3.
Process Affected	A description of the process to be affected is given in chapter 3.
Availability of Control Technology	Information on the availability of control technology is given in chapter 4.
Existing Regulations at State or Local Level	A discussion of existing regulations on the industry to be affected by the standard is included in chapter 3, section 2.3.5.
2. Alternatives to the Proposed Action	
a. Postponing Action	
Environmental Impacts	Environmental effects of delaying the standards are discussed in chapters 6 and 8.
b. Dry Control Alternative	
Environmental Impacts	The environmental impacts associated with this alternative are discussed in chapter 6.
Costs	The costs of a dry control alternative are discussed in chapter 7.
3. Environmental Impact of Proposed Action	
Air Pollution	The air pollution impact of the standards is considered in chapter 6, section 6.1.

Appendix B (continued)

Agency Guidelines for Preparing Regulatory Action Environmental Impact Statements (39 FR 37419)	Location Within the Standards Support and Environmental Impact Statement
Water Pollution	The water pollution impact of the standards is discussed in chapter 6, section 6.2
Solid Waste Disposal	The solid waste disposal impact of the standards is discussed in chapter 6, section 6.3.
Energy	The energy impact of the standards is considered in chapter 6, section 6.4.

APPENDIX C. EMISSION TEST DATA SUMMARY

C.1 INTRODUCTION

This appendix summarizes the emission source test data cited in the main body of this document. It describes the tested facilities (gas turbine size, type, operating conditions, and the exhaust gas stream characteristics), reviews the testing methods used and summarizes the results of the emissions measurement tests.

Facilities are identified by the same coding used in the main body of the document. For example, Table 3 summarizes the emission source test data obtained from facility C.

A variety of sampling and analytical techniques were used to develop the data contained in this appendix. Each of these techniques is discussed briefly in the next section.

During the development of standards of performance for stationary gas turbines, the major gas turbine manufacturing companies in the United States were contacted, along with those electric utility companies known to have experience with the control of emissions from gas turbines. Those companies which manufacture most of the turbines sold in this country were then visited as were a number of the electric utility companies who operate gas turbines.

An evaluation of emission control technology (as it applies to gas turbines) was also undertaken by means of a literature survey, personal contacts and plant visits. A great deal of research on combustion modifications is underway and, as discussed in Chapter 4, has already demonstrated considerable success in reducing NO_x emissions as evidenced by rig tests and tests of prototype engines. These combustion modifications, however, have not yet been applied to production engines to the extent of achieving reductions in NO_x emissions as great as those that have been demonstrated through the injection of water or steam directly into the gas turbine combustors. EPA tested three turbines operated by one utility which used water or steam injection. The data listed in the tables of section C.5 are emission source test data reported to us by utility companies, turbine manufacturers, and local pollution control agencies, as well as the results of the EPA tests.

C.2 DESCRIPTION OF EMISSION TEST METHODS

The test methods used to obtain the emission data summarized in this appendix are summarized below in two parts. The sampling procedures have been identified by letter codes and the analysis techniques by numbers. A test method can consist of any letter-digit combination. Test method A1, for example, consists of sampling method A and analysis method 1.

SAMPLING PROCEDURES

A. A movable probe assembly was used for sampling. The assembly is comprised of two adjustable single point probes, one to survey the vertical and the other to survey the horizontal radii of the engine tailpipe. Each probe is center-positioned in each of five equal annulus areas with a total of 10 points being sampled.

B. A fixed probe technique was used for sampling. The probe incorporated three sampling points in each of four parallel branches which were manifolded to a single sample line.

C. The sampling technique described in the Los Angeles Air Pollution Control District Source Testing Manual was used.

D. Two fixed probes located at right angles to each other and perpendicular to the gas stream. Each probe contained 11 sampling ports. Sample lines were manifolded together to provide an average gas sample.

E. Three two-litre grab samples were taken during each test for NO_x determination with continuous sampling for CO_2 and O_2.

F. The single point sampling techniques described in the Los Angeles Air Pollution Control District Source Testing Manual were used.

G. Samples were obtained by traversing the gas turbine stack with a single port probe. The number of sampling points was selected in accordance with EPA Method 1, entitled, "Sampling and Velocity Traverses for Stationary Sources", which was published December 23, 1971.

H. Samples were extracted at four pressure probes located between the compressor and power turbines. Sample lines were manifolded together to provide an average gas sample.

J. Three stationary probes were located at each of two sampling stations, one upstream of the exhaust silencing baffles and the other downstream.

K. A fixed, single-point probe was used for sampling.

L. Two gas-sampling techniques were used, an annular traversing probe and a fixed sampling rake. The traversing probe contained four radial sampling ports which were normally manifolded together. The probe was centered in each of five equal annulus areas with a total of 20 points being sampled. The fixed rake contained five sampling probes, each containing four radial sampling ports connected to a single line. The five probes (20 sampling points) were normally manifolded together to provide a single sample.

M. A traversing rake was used for sampling. It was comprised of five probes which radiated out from the engine exhaust center line and was rotated through 12 traverse positions for each test. Each probe contained five ports, of which three were normally used for gas sampling - for a total of 180 samples per traverse. Usually, the lines from each port in a probe were manifolded together to provide an average gas sample for each probe.

N. Samples were obtained by sampling at seven radial locations around the stack using a five port probe, for a total of 35 sampling ports. Sample lines from each port were manifolded together to provide an average sample for each probe position. The number of sampling points was determined in accordance with EPA Method 1, entitled, "Sampling and Velocity Traverses for Stationary Sources", which was published December 23, 1971.

ANALYSIS TECHNIQUES

1. Chemiluminescent analyzer with a thermal converter was used to convert NO_2 to NO for total NO_x, a non-dispersive infrared detector (NDIR) was used for CO and CO_2 and a flame ionization detector (FID) measured HC emissions.
2. The phenoldisulphonic acid (PDS) and Saltzman methods were used for analysis of NO_x, a gas chromatograph (GC) was used for CO determination and an Orsat analyzer was used to measure O_2 and CO_2 concentrations.

3. The PDS and Saltzman methods were used for analysis of NO_x, CO was estimated with a Mine Safety Appliance (MSA) indicator tube, hydrocarbons (HC) levels were determined with an FID and O_2 and O_2 were measured with an Orsat analyzer.
4. PDS and electrochemical methods were used to determine NO_x concentrations, an NDIR measured CO and HC emissions, O_2 was measured by Orsat and with a process monitor, and CO_2 was measured by Orsat.
5. PDS and electrochemical techniques were used for analysis of NO_x. An electrochemical analyzer was used for CO determination, an FID measured HC in the exhaust gases, O_2 and CO_2 levels were determined using GC techniques and smoke was measured by ASTM Smoke Spot method D-2156.
6. NDIR and Ultraviolet Analyses were used to measure NO, and NO_2, and NDIR was used to measure CO. O_2 was calculated from the fuel air ratio.
7. NO_x was determined using PDS, electrochemical, chemiluminescent and integrated lead dioxide techniques, CO was measured by NDIR, HC by FID, and O_2 by polargraphic and paramagnetic techniques.

C.3 DESCRIPTION OF FACILITIES

A. A single shaft gas turbine with a rated output of 0.03 megawatt (40 hp). The production (A1) turbine had no emissions controls. The development (A2) turbine utilized vaporizing combustors for emissions control. Test Method A1 was used and data were provided by the manufacturer. The turbine was tested in idle mode and at rated output. Fuel used was aviation kerosene.

B. A single-shaft gas turbine with a rated output equivalent to 0.158 megawatts (212.5 hp). These turbines had no emissions controls. They were tested over the operating load range using method A1 and data were provided by the manufacturer. The gas turbines in Test series 1 were operated on Jet A-1 fuel and the unit in test 2 was operated on aviation kerosene.

C. An advanced engine test rig for a single-shaft gas turbine with an output of about 0.158 megawatt (212.5 hp). A number of dry control techniques were tested over the operating load range, singly and in combination, using test method A1 and data were provided by the manufacturer. Units were operated on aviation kerosene and natural gas during the testing.

D. A combustor test rig for a single-shaft gas turbine with a rated output equivalent to 0.158 megawatt (212.5 hp). This combustor was tested without NO_x controls and also with staged fuel combustion. It was tested over the operating load range using method A1 and data were supplied by manufacturer. Fuel used for these tests was not specified, but was probably aviation kerosene.

E. A single-shaft gas turbine with a rated output of 0.25 megawatt. It had no emissions controls. Test method B1 was used and data were provided by the manufacturer. Operation was at rated output and the fuel used was aviation kerosene.

F. A regenerative-cycle two-shaft gas turbine with a rated base load output of 0.20 megawatts. This turbine incorporates a short residence time primary zone to reduce NO_x formation. The test method is unknown. The data were supplied by the manufacturer and are for operation at rated output. Fuels used were #2 distillate and natural gas.

G. A single-shaft aircraft turboprop engine which is aerodynamically identical to facility H. The G1 engine has a rated shaft output of 665 horsepower (.497 MW), had no NO_x controls, and was tested over the operating load range using method A1. The G2 engine is rated at 690 horsepower (.51 MW) and was tested at 100% of design capacity with water injection, also using method A1. Data for G1 was developed for EPA by the manufacturer and G2 data was supplied by the manufacturer. G1 tests 1 and 2 were run with aviation kerosene and JP5, respectively. The fuels used for most of the G2 tests were not specified.

H. A single-shaft industrial gas turbine with a rated output of 0.51 megawatts (690 hp). The H1 engine was tested over the operating load range using method A1 and data was provided by the manufacturer. The H1 test data represents an average of six tests on units operating on natural gas (test series 1) and 11 on DF-2 (test series 2). For H2, a number of dry control techniques were tested, singly and in combination, during operation on DF-2.

J. A combustor test rig for a single-shaft gas turbine with a rated output of 0.51 megawatts (690 hp). A number of dry control techniques were tested, singly and in combination, over the operating load range. Test method A1 was used and data was provided by the manufacturer. Fuel used was DF-2.

K. A single-shaft or two-shaft gas turbine with a rated output of 1100 horsepower (0.75 - 0.80 MW continuous output). These turbines had no emissions controls. Facility K1 was tested over the operating load range using test method C1 plus polargraphic analysis for O_2 and chemical absorption for CO_2 and O_2. The data were provided by a user. Facility K2 test series, 15 units tested on kerosene (test 1) and 9 units tested on natural gas (test 2), were conducted at rated output using test method B1, and the data were provided by the manufacturer.

L. Single-shaft gas turbine with a rated output of 1.0 megawatts. This turbine had no emissions controls. The test method is unknown. It was tested over the operating load range on #2 distillate fuel and data were provided by the manufacturer.

M. Single-shaft or two-shaft gas turbines with a rated peak output of 2.5 MW, or 2710 hp continuous. Three turbines were tested on kerosene (test series 1) and 8 on natural gas (test series 2). They had no emissions controls. The turbines were tested at rated capacity using test method M1, and data were provided by the manufacturer.

N. A production prototype gas turbine with a rated output of 2.5 megawatts. It had no NO_x controls but did use an air atomizing combustor for smoke reduction. It was tested over the operating range using test method M1 and data were provided by the manufacturer. Fuels used were natural gas, kerosene and #2 distillate oil.

O. A regenerative cycle two-shaft development turbine with a rated output of 2.5 megawatts. This turbine utilized a lean primary zone for NO_x control. It was tested at no load on natural gas and at rated output on all fuels. Test method M1 was used and data were provided by the manufacturer. Fuels used were natural gas, kerosene, naptha and #2 distillate oil.

P. A single-shaft gas turbine with a rated output of 2.5 megawatts. Facility P1 had no NO_x controls but the combustor and fuel nozzle had been modified for smoke and particulate emissions control. It was tested at no load on #1 distillate oil and at rated load on both #1 distillate and natural gas. The test method used is unknown and data were provided by the manufacturer. Facility P2 was tested before and after combustion modifications were made to reduce particulate emissions. It was tested at no load and at rated output on distillate fuel using test method C1 and data were provided by a user.

Chemical absorption was also used for CO_2 and O_2 measurement, and polargraphic analysis for O_2.

Q. A combustor test rig for a single-shaft gas turbine with a rated output of 2.5 megawatts. Water injection was used for NO_x control. Water was injected into the inlet air stream of a production combustor (test 1) or into the combustor itself (test 2) at several flow rates during operation at full rated load. The test method used is unknown. Test data were provided by the manufacturer. Fuel used was #1 distillate.

R. A combustor test rig for a single-shaft gas turbine with a rated output of 2.5 megawatts. An R & D combustor was equipped with variable geometry and staged combustion for NO_x control. It was tested at no load and at rated output on # distillate fuel. The test method used is unknown. Data were provided by the manufacturer.

S. A gas turbine with a base load rating of 6.0 megawatts (7.5 megawatts peak). Prevaporization of the liquid fuel is utilized to reduce emissions. The test method is unknown, except that PDS was used for NO_x analysis. Data were provided by the manufacturer and are for the turbine operating on distillate oil at rated base load capacity and on natural gas and distillate oil at peak load output.

T. A two-shaft, free-turbine (aircraft type) gas turbine with a rated output of 10.3 megawatts (13,900 horsepower). It had no NO_x emissions controls. Facility T1 was tested at rated output on natural gas and distillate oil and data were provided by a user. The test method is unknown. Facility T2 was tested over the load range using method D1 (less CO_2 measurment) and data were provided by the manufacturer. Fuels used for both series of tests were natural gas and distillate oil.

U. A two-shaft, free turbine (aircraft type) gas turbine with a rated peak load output of 13 megawatts (13 MW base). Facility U3 had no NO_x control. Water injection was used for NO_x control on Facilities U1 and U2. Water was injected into the combustor at several flow rates during operation at rated and peak loads. Test method F2 was used and data were provided by a user. Fuel used was natural gas and, for one test, #2 distillate oil.

V. A single-shaft gas turbine with a rated peak load output of 17.2 megawatts. Facilities V1 and V2 were equipped with an atomizing air-fuel injection system to reduce visible emissions and utilized water injection for NO_x control. Tests were run at spinning reserve and at about peak output using natural gas and #2 distillate oil. Facility V1 was tested using test method E3 and data were provided by a user. Facility V2 testing was accomplished using test method G7, with some tests performed using test method F7. Facility V3 was tested with no controls, with dry controls, with water injection, and with a combination of wet and dry controls. Tests were run at peak output using natural gas and #2 distillate oil. Test method F4 was used and data were provided by the manufacturer.

W. A two-shaft, free-turbine gas turbine with a rated maximum peak output of 21.3 megawatts. Steam injection was used for NO_x control. Most of the tests in this turbine used test method G7, with a few tests using test method F7. Tests were run in the spinning reserve mode and at peak output. The fuel used was JP-5.

X. A two-shaft gas turbine with a rated peak output of 27.6 megawatts. It had no emissions controls. Test method H6 was used plus Orsat for CO_2, and data was provided by the manufacturer. Fuels used were distillate oil and methanol. The turbine was tested at about 70% of rated capacity, and results on methanol fuel were extrapolated to the full load point. The fuel delivery system was sized for distillate, which has about twice the heat of combustion of methanol, so full power output could not be achieved with methanol fuel.

Y. A single-shaft gas turbine with a rated peak output of 32.8 megawatts. Water injection was used for NO_x control. This turbine was tested using test method J5 and data were provided by manufacturer. Tests were run at low load and at rated capacity using #2 distillate oil.

Z. A combustor test rig for a single-shaft gas turbine with an unspecified power output. A number of dry control techniuqes and water injection were utilized for NO_x control, singly and in combination, and tests were run at rated output. Conventional combustors were also tested, for reference. The sampling method used is unknown, but analysis method 5 was used. Data were provided by the manufacturer. Facility Z1 and Z2 were both tested on #2 distillate oil and facility Z1 was also tested on natural gas.

AA. A scaled combustor test rig for a single-shaft gas turbine of unspecified output. Various dry control techniques and steam injection were utilized for NO_x control. Conventional combustors were also tested, for reference. Tests were run at rated output using test method K1. Also, NO_x was measured by NDIR and smoke by ASTM D-2156. Data were supplied by the manufacturer. Fuels used were natural gas and #2 distillate oil.

BA. A scaled combustor test rig for a single-shaft gas turbine of unspecified output. Catalytically-supported combustion was used for NO_x control, with a conventional combustor and a premix combustor included for comparison. Facility BA1 was tested using test method K1. Also, NO_x was measured by NDIR and smoke by ASTM D-2156. Data were supplied by the manufacturer. Facility BA2 was tested with an unknown sampling method and analysis method 1, and data were provided by control device manufacturer. A variety of fuels were used in the tests, including synthetic coal gas and nitrogen-doped propane and distillate.

CA. An engine test rig for a dual spool axial flow turbofan aircraft engine producing 44,300 pounds of thrust at take-off. A number of dry control techniques were utilized for NO_x control and a conventional combustor was included for comparison purposes. Test method L6 was used plus NDIR for CO_2 and SAE ARP 1179 for smoke. Data were developed by the manufacturer for NASA and tests were run at rated output on Jet-A fuel.

DA. A combustor test rig for a dual rotor high bypass ratio turbofan aircraft engine producing about 50,000 pounds of thrust at take-off. Variable geometry and exhaust gas circulation were utilized for NO_x control. Conventional combustor results are also included for reference. Test method M1 was used, less HC measurement and plus SAE A1179 for smoke. Data were developed by the manufacturer for NASA. Tests were run over the load range and are reported in the data summary for rated output (take-off) and 69%, which is assumed equivalent to base load for a stationary gas turbine. Fuel used was Jet-A.

EA. A single-shaft gas turbine with a rated base load output of 26 megawatts. It had no NO_x controls. The sampling procedure is unknown, but analysis method 5 was used and data was provided by the manufacturer. The turbine was tested over the load range (up to 85% of base load) to determine the effect of different fuels on emissions. Fuels used were natural gas, #2 distillate, and a heavy distillate.

FA. A single-shaft simple cycle gas turbine with a rated output at base load of 51.7 megawatts. Water injection was used for NO_x control. Test method N1 was used (less hydrocarbon measurement) and O_2 was measured using paramagnetic techniques. The turbine was tested over the load range and data were supplied by the manufacturer. Fuel used was #2 distillate.

GA. A single-shaft gas turbine with a rated output of 52.9 megawatts. It had no NO_x controls. The sampling method is not known, but analysis method 1 was used and data were provided by the manufacturer. The turbine was tested at rated output on #2 distillate and on crude oil.

HA. A single-shaft simple cycle gas turbine with a rated base load output of 61.5 megawatts on natural gas and 60.4 megawatts on #2 distillate fuel. Water injection was used for NO_x control. It was tested on distillate oil (HA1) and natural gas (HA2) at several water flow rates during operation at close to rated output. The sampling method is unknown, but analysis method 1 was used, except that CO_2 was not measured. Data were provided by the manufacturer.

JA. A single-shaft regenerative cycle gas turbine with a rated base load output of 58.6 megawatts. It had no NO_x controls. The sampling method is unknown, but analysis method 1 was used. Also, O_2 was measured using paramagnetic techniques. Tests were run at close to rated output on #2 distillate oil, and data were supplied by the manufacturer.

C.4 TEST DATA CODING SYSTEM

NO_x Controls

A. Combustion Modification (Dry Controls)

1. Lean Primary Zone
   - a. 45% Primary air
   - b. 60% Primary air (no secondary air)
2. Premix/Lean premix/Staged premix
3. Reduced residence time
4. Variable geometry
5. Exhaust gas recirculation
6. Staged fuel injection

    7. Radial/axial fuel staging
    8. Airblast/piloted airblast
    9. Vaporizing combustor/prevaporizing chamber
    10. Pressure atomizer
    11. Lean dome double annular
    12. Staged combustion
    13. Swirl can
    14. Swirl vorbix

B. Water Injection

    1. Water added in combustor primary zone.
       M. Methanol added to water.
    2. Water added ahead of combustor.

C. Combustor Modification Plus Water Injection

D. Steam Injection

    1. Water added in combustor primary zone.
    2. Water added ahead of combustor.

E. Catalytically Supported Combustion

FUELS

DF-1	#1 distillate fuel
DF-2	#2 distillate fuel
Jet-A	Aviation kerosene
JP-5	Jet Kerosene
K	Kerosene
M	Methanol
P	Propane
NG	Natural gas
CG	Coal gas
SCG	Synthetic coal gas

Efficiency Correction Factor (EF)

The efficiency factor (EF) is applied to the NO_x emissions concentration (corrected to 15% O_2) using the following formula:

$$X = \frac{X_1 \ (Y)}{10180}$$

Where:

X = NO_x concentration (@ 15% O_2) after correction factor applied

X_1 = NO_x concentration @ 15% O_2

Y = Heat rate of test unit, in Btu/hph

The purpose of this formula is to correct for the change in O_2 content of the exhaust gas stream which is directly attributable to the greater efficiency of the test unit, compared to the reference unit (which has a heat rate of 10,180 Btu/hph).

TABLE 1

Facility A1 and A2

Summary of Results

Unit Type	Production Gas Turbine				Development Gas Turbin			
Test Number	1A		1B		2A		2B	
Test Date	5-17-71		5-18-71		--		--	
% Rated Output (0.03 mw)	0	102.8	0	102.4	0	100	0	100
NO_x Controls	None		A-9		None		A-9	
Water-Fuel Ratio								
Fuel	Jet A							
Stack Effluent								
Flow rate - lb/sec								
Temperature - °F	640	795	610	850				
Water vapor - Vol. %								
CO_2 - Vol. % (wet)	1.72	2.32	1.74	2.20				
O_2 - Vol. % (dry)		17.9		18.0				
Visible Emissions - % opacity								
Nitrogen Oxides Emissions (as NO_2)								
ppm (dry)	12.7	21.6	19.2	26.2				
ppm @ 15% O_2		42		53	28.8	36.9	43.5	46.9
ppm @ 15% O_2 w/EF(1)		42		53	28.8	36.9	43.5	46.9
lb/hr	0.84	0.145	1.31	0.195				
Carbon Monoxide Emissions								
ppm (dry)	331	170	235	333				
ppm @ 15% O_2		347		666	751	291	536	595
lb/hr	1.34	0.69	0.97	1.50				
Hydrocarbon Emissions								
ppm (dry)								
ppm @ 15% O_2					119	11.3	16.5	11.6
lb/hr	.121	.015	.017	.017				

NOTES: (1) Efficiency factor of no benefit.

TABLE 2
Facility B
Summary of Results

Unit Type	Production Gas Turbine			
Test Number	1A(1)	1B(1)	2A(2)	2B(2)
Test Date	9/73 thru 3/75		5/75	
% Rated Output (0.158 equiv. mw)	0	90	0	90
NO_x Controls	None		None	
Water-Fuel Ratio				
Fuel	Jet-A		Jet-A	
Stack Effluent				
Flow Rate - lb/sec	4.14	3.08	4.18	3.14
Temperature - °F	534	1163	525	1160
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry)				
Visible Emissions - % opacity				
- SAE smoke	45.3	33.8	38.2	29.7
Nitrogen Oxides Emissions				
ppm (dry)				
ppm @ 15% O_2	46.1	54.7	50.2	57.6
ppm @ 15% O_2 @ EF(3)	46.1	54.7	50.2	57.6
lb/hr	0.37	0.93	0.39	0.97
Carbon Monoxide Emissions				
ppm (dry)				
ppm @ 15% O_2	1282	323	1163	306
lb/hr	6.26	3.36	5.46	3.14
Hydrocarbon Emissions				
ppm (dry)				
ppm @ 15% O_2	469	6.4	5.46	3.3
lb/hr	1.30	0.04	1.47	0.02

TABLE 2
Facility B

NOTES:

1. Average of tests on 6 units
2. Single tests
3. Efficiency factor of no benefit

TABLE 3
Facility C
Summary of Results

Unit Type	Advanced Engine Test Rig							
Test Number	1A	1B	2A	2B	3A	3B	3C	3D
Test Date								
% Rated Output (0.158 mw)(2)	0	100	0	100	0	100	0	100
NO_x Controls	None		A-8		A-2, A-4 & A-9			
Water-Fuel Ratio								
Fuel	Jet-A		Jet-A		Jet-A		N.G.	
Stack Effluent								
Flow rate - lb/sec								
Temperature - °F								
Water vapor - Vol. %								
CO_2 - Vol. % (dry)								
O_2 - Vol. % (dry)								
Visible Emissions - % opacity								
Nitrogen Oxides Emissions								
ppm (dry)								
ppm @ 15% O_2	223	--	182	199	136	180	104	140
ppm @ 15% O_2 @ EF(1)	223	--	182	199	136	180	104	140
lb/hr								
Carbon Monoxide Emissions								
ppm (dry)								
ppm @ 15% O_2	44	--	93	36.6	361	120	184	65.9
lb/hr								
Hydrocarbon Emissions								
ppm (dry)								
ppm @ 15% O_2	3.6	--	3.5	13.6	16.9	6.4	11.9	4.9

Notes:

1. Efficiency factor of no benefit
2. Estimated output of engine not necessarily of test rig

TABLE 4
Facility D
Summary of Results

Unit Type	Combustor Test Rig			
Test Number	1A	1B	2A	2B
Test Date				
% Rated Output (.158 mw)(2)	0	100	0	100
NO_x Controls	None		A-6	
Water-Fuel Ratio				
Fuel	UNK		UNK	
Stack Effluent				
Flow rate - lb/sec	1.11	1.14	1.11	1.14
Temperature - ^oF				
Water vapor - Vol. %				
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry)				
Visible Emissions - % opacity				
Nitrogen Oxides Emissions				
ppm (dry)				
ppm @ 15% O_2	64.4	57.4	74.3	54.7
ppm @ 15% O_2 @ EF(1)	64.4	57.4	74.3	54.7
lb/hr				
Carbon Monoxide Emissions				
ppm (dry)				
ppm @ 15% O_2	453	327	1286	537
lb/hr				
Hydrocarbon Emissions				
ppm (dry)				
ppm @ 15% O_2	5.9	3.0	405	8.0
lb/hr				

Notes:

1. Efficiency factor of no benefit.
2. Output of engine, not necessarily test rig.

TABLE 5

Facility E

Summary of Results

Unit Type	Production Gas Turbine
Test Number	1(1)
Test Date	9/72 - 8/73
% Rated Output (0.20 mw)	100
NO_x Controls	None
Water-Fuel Ratio	
Fuel	Kerosene
Stack Effluent	
Flow rate - lb/sec	
Temperature - ^oF	
Water vapor - Vol. %	
CO_2 - Vol. % (dry)	
O_2 - Vol. % (dry)	
Visible Emissions - % opacity	< 5
Nitrogen Oxides Emissions	
ppm (dry)	
ppm @ 15% O_2	56
ppm @ 15% O_2 @ EF(2)	56
lb/hr	
Carbon Monoxide Emissions	
ppm (dry)	
ppm @ 15% O_2	250
lb/hr	
Hydrocarbon Emissions	
ppm (dry)	
ppm @ 15% O_2	
lb/hr	

Notes:

1. Data represents average of tests on 9 units
2. Corrector factor of no benefit

TABLE 6

Facility F

Summary of Results

Unit Type	Regenerative Cycle Stationary Gas Turbine	
Test Number	1	2
Test Date		
% Rated Output (0.20 mw)	100	100
NO_x Controls (Dry)	A-3	A-3
Water-Fuel Ratio		
Fuel	DF-2	N.G.
Stack Effluent		
Flow rate - lb/sec		
Temperature - ^oF		
Water vapor - Vol. %		
CO_2 - Vol. % (dry)		
O_2 - Vol. % (dry)	17.7	17.7
Visible Emissions - % opacity	2	2 (est.)
Nitrogen Oxides Emissions		
ppm (dry)	115	62
ppm @ 15% O_2	209	113
ppm @ 15% O_2 @ EF	173	94
lb/hr		
Carbon Monoxide Emissions		
ppm (dry)	82	38
ppm @ 15% O_2	149	70
lb/hr		
Hydrocarbon Emissions		
ppm (dry)		
ppm @ 15% O_2		
lb/hr		

TABLE 7
Facility G1
Summary of Results

Unit Type	Production Aircraft Turbo Prop Engine			
Test Number	1A	1B	2A	2B
Test Date	6/1/71		4/30/71	
% Rated Output (.497 mw)	0.2	91.2	0.6	90.6
NO_x Controls	None		None	
Water-Fuel Ratio				
Fuel	Jet-A		JP5	
Stack Effluent				
Flow rate - lb/sec				
Temperature - ^oF	735	1050	--	1080
Water vapor - Vol. %				
CO_2 - Vol. % (dry)	--	3.45	--	3.64
O_2 - Vol. % (dry)	--	16.4		16.1
Visible Emissions - % opacity				
Nitrogen Oxides Emissions				
ppm (dry)	3.1	66.5	16.6	119
ppm @ 15% O_2	--	88	--	147
ppm @ 15% O_2 @ EF[1]	--	88	--	147
lb/hr	0.15	2.30	0.24	4.00
Carbon Monoxide Emissions				
ppm (dry)	388	16.4	381	34.8
ppm @ 15% O_2	--	21	--	43
lb/hr	3.19	0.35	3.34	0.71
Hydrocarbon Emissions				
ppm (dry)	549	5.9	99	7.6
ppm @ 15% O_2	--	7.7	--	9.3
lb/hr	2.58	.071	0.31	0.05

Notes:

1. Efficiency factor of no benefit

TABLE 8

Facility G2

Summary of Results

Unit Type	Aircraft Turbo Prop Engine						
Test Number	1A	1B	1C	2A	2B	3A	3B
Test Date							
% Rated Output (.51 mw)		100		100		100	
NO_x Controls		B-2		B-1		B-1M	
Water-Fuel Ratio	0.5	1.0	1.5	0.5	1.0	0	0.5
Fuel		UNK		UNK		Jet A	
Stack Effluent							
Flow rate - lb/sec							
Temperature - °F							
Water vapor - Vol. %							
CO_2 - Vol. % (dry)							
O_2 - Vol. % (dry)							
Visible Emissions - % opacity							
% change	+40	+45	+48	-23	-34		
Nitrogen Oxides Emissions							
ppm (dry)							
ppm @ 15% O_2						148	87.6
ppm @ 15% O_2 @ EF(1)						148	87.6
lb/hr							
percent change	-41	-41	-50	-36	-60	N/A	40
Carbon Monoxide Emissions							
ppm (dry)							
ppm @ 15% O_2						17.3	1069
lb/hr							
percent change							
Hydrocarbon Emissions							
ppm (dry)							
ppm @ 15% O_2						8.5	489
lb/hr							
percent change	negl.	negl.	negl.	0	0	--	--

Note:

1. Efficiency factor of no benefit.

TABLE 9
Facility H1
Summary of Results

Unit Type	Production Industrial Engine			
Test Number	1A(1)	1B(1)	2A(2)	2B(2)
Test Date	1/21 thru 5/12/75		10/74 thru 3/75	
% Rated Output (0.51 mw)	0	97.3	0	90
NO_x Controls	None		None	
Water-Fuel Ratio				
Fuel	N.G.		DF 2	
Stack Effluent				
Flow rate - lb/sec				
Temperature - ^{o}F	518	919	521	876
Water vapor - Vol. %				
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry)				
Visible Emissions - % opacity				
- SAE smoke			19	27
Nitrogen Oxides Emissions				
ppm (dry)				
ppm @ 15% O_2	102	109	125	149
ppm @ 15% O_2 @ EF(3)	102	109	125	149
lb/hr	1.46	3.69	1.77	4.76
Carbon Monoxide Emissions				
ppm (dry)				
ppm @ 15% O_2	119	7.0	147	21.1
lb/hr	1.05	0.15	1.27	0.41
Hydrocarbon Emissions				
ppm (dry)				
ppm @ 15% O_2	24.6	4.2	35.5	17.8
lb/hr	0.12	.05	0.18	0.20

Notes:

1. Average of tests on 6 units
2. Average of tests on 11 units
3. Efficiency factor of no benefit.

TABLE 10

Facility H2

Summary of Results

Unit Type	Production Industrial Engine							
Test Number	1A	1B	2A	2B	3A	3B	4A	4B
Test Date								
% Rated Output (0.51 mw)	0	100	0	100	0	100	0	100
NO_x Controls	None		A-1		A-8		A-1 & A-8	
Water-Fuel Ratio								
Fuel	DF-2		DF-2		DF-2		DF-2	
Stack Effluent								
Flow rate - lb/sec								
Temperature - ^oF								
Water vapor - Vol. %								
CO_2 - Vol. % (dry)								
O_2 - Vol. % (dry)								
Visible Emissions - % opacity								
Nitrogen Oxides Emissions								
ppm (dry)								
ppm @ 15% O_2	136	157	127	126	175	164	133	139
ppm @ 15% O_2 @ EF(1)	136	157	127	126	175	164	133	139
lb/hr								
Carbon Monoxide Emissions								
ppm (dry)								
ppm @ 15% O_2	130	16.2	336	51.2	93.8	22.5	231	44.8
lb/hr								
Hydrocarbon Emissions								
ppm (dry)								
ppm @ 15% O_2								
lb/hr								

Notes:

1. Efficiency factor of no benefit

TABLE 13
Facility K2
Summary of Results

Unit Type	Production 1 & 2 Shaft Gas Turbines	
Test Number	1(1)	2(2)
Test Date	9/72-8/73	9/72-8/73
% Rated Output (0.80 mw)	100	100
NO_x Controls	None	None
Water-Fuel Ratio		
Fuel	Kerosene	Natural Gas
Stack Effluent		
Flow rate - lb/sec		
Temperature - ^{o}F	860	860
Water vapor - Vo.. %		
CO_2 - Vol. % (dry)		
O_2 - Vol. % (dry)	16.5	17.0
Visible Emissions - % opacity	<10	<10
Nitrogen Oxides Emissions		
ppm (dry)	52	34
ppm @ 15% O_2	69	51
ppm @ 15% O_2 @ EF	69	51
lb/hr		
Carbon Monoxide Emissions		
ppm (dry)	122	45
ppm @ 15% O_2	163	68
lb/hr		
Hydrocarbon Emissions		
ppm (dry)		
ppm @ 15% O_2		
lb/hr		

Notes:

1. Data represents average of tests on 15 units.
2. Data represents average of tests on 9 units.
3. Efficiency factor of no benefit.

TABLE 11
Facility J
Summary of Results

Unit Type	Combustion Rig							
Test Number	1A	1B	2A	2B	3A	3B	4A	4B
Test Date								
% Rated Output (0.51 mw)$^{(2)}$	0	100	0	100	0	100	0	100
NO_x Controls	None		A-8		A-1		A-1 & A-8	
Water-Fuel Ratio								
Fuel	DF-2		DF-2		DF-2		DF-2	
Stack Effluent								
Flow Rate - lb/sec								
Temperature - ^{o}F								
Water vapor - Vol. %								
CO_2 - Vol. % (dry)								
O_2 - Vol. % (dry)								
Visible Emissions - % opacity								
Nitrogen Oxides Emissions								
ppm (dry)								
ppm @ 15% O_2	108	156	170	160	139	124	132	132
ppm @ 15% O_2 @ EF$^{(1)}$	108	156	170	160	139	124	132	132
lb/hr								
Carbon Monoxide Emissions								
ppm (dry)								
ppm @ 15% O_2	248	17.7	99.8	11.7	429	33.0	231	22.2
lb/hr								
Hydrocarbon Emissions								
ppm (dry)								
ppm @ 15% O_2	9.9	1.0	--	--	26.8	2.4	--	--
lb/hr								

NOTES:

1. Efficiency factor of no benefit.
2. Output of engine, not necessarily of test rig.

TABLE 12
Facility K1
Summary of Results

Unit Type	Production	
Test Number	1A	1B
Test Date	2/72 - 4/72	
% Rated Output (0.75 mw)	0	100
NO_x Controls	None	
Water-Fuel Ratio		
Fuel	DF	
Stack Effluent		
Flow rate - lb/sec		
- SCFM	11,300	11,000
Temperature - ^{o}F	476	850
Water vapor - Vol. %		
CO_2 - Vol. % (dry)	0.5	3.0
O_2 - Vol. % (dry)	19.0	16.5
Visible Emissions - % opacity		
Nitrogen Oxides Emissions		
ppm (dry)	3.0	14.5
ppm @ 15% O_2	9.0	19.3
ppm @ 15% O_2 @ EF(1)	9.0	19.3
lb/hr		
Carbon Monoxide Emissions		
ppm (dry)	139	90
ppm @ 15% O_2	390	120
lb/hr		
Hydrocarbon Emissions		
ppm (dry)	14.0	9.0
ppm @ 15% O_2	42.0	12.0
lb/hr		

Notes:

1. Efficiency factor of no benefit.

TABLE 14
Facility L
Summary of Results

Unit Type Radial Industrial Gas Turbine Generating Set

Test Number	1A	1B
Test Date		
% Rated Output (1.0 mw)	0	100
NO_x Controls	None	None
Water-Fuel Ratio		
Fuel	DF-2	DF-2
Stack Effluent		
Flow rate - lb/sec		
Temperature - ^oF		
Water vapor - Vol. %		
CO_2 - Vol. % (dry)		
O_2 - Vol. % (dry)	19.0	16.2
Visible Emissions - % opacity	0	0
Nitrogen Oxides Emissions		
ppm (dry)	30	52
ppm @ 15% O_2	90	69
ppm @ 15% O_2 @ EF(1)	90	69
lb/hr	4.6	7.4
Carbon Monoxide Emissions		
ppm (dry)	100	40
ppm @ 15% O_2	300	54
lb/hr	9.4	3.5
Hydrocarbon Emissions		
ppm (dry)	<10	<10
ppm @ 15% O_2	--	--
lb/hr	--	--

Notes:
1. Efficiency factor of no benefit

TABLE 15

Facility M

Summary of Results

Unit Type	Production 1 & 2 Shaft Gas Turbines	
Test Number	$1^{(1)}$	$2^{(2)}$
Test Date	9/72-8/73	9/72-8/73
% Rated Output (2.5 mw)	100	100
NO_x Controls	None	None
Water-Fuel Ratio		
Fuel	Kerosene	Natural Gas
Stack Effluent		
Flow rate - lb/sec		
Temperature - ^oF	800	800
Water vapor - Vol. %		
CO_2 - Vol. % (dry)		
O_2 - Vol. % (dry)	16.5%	17.0%
Visible Emissions - % opacity	>10	<10
Nitrogen Oxides Emissions		
ppm (dry)	65	59
ppm @ 15% O_2	86	89
ppm @ 15% O_2 @ EF	82	85
lb/hr		
Carbon Monoxide Emissions		
ppm (dry)	95	45
ppm @ 15% O	126	68
lb/hr		
Hydrocarbon Emissions		
ppm (dry)		
ppm @ 15% O_2		
lb/hr		

Notes:
1. Data represents average of tests on 3 units.
2. " " " " " " 8 "

TABLE 16
Facility N
Summary of Results

Unit Type	Production Prototype Gas Turbine					
Test Number	1A	1B	2A	2B	3A	3B
Test Date	5/3/75	5/3/75	5/3/75	5/3/75	5/3/75	5/5/75
% Rated Output (2.5 mw)	0	100	0	100	0	100
NO_x Controls	None(1)					
Water-Fuel Ratio						
Fuel	N.G.	N.G.	Kero	Kero	DF-2	DF-2
Stack Effluent						
Flow rate - lb/sec						
Temperature - ^oF						
Water vapro - Vol. %						
CO_2 - Vol. % (dry)	0.93	2.36	1.18	2.92	1.22	2.99
O_2 - Vol. % (dry)						
Visible Emissions - % opacity			0.4	3.0	0	3.8
Nitrogen Oxides Emissions						
ppm (dry)						
ppm @ 15% O_2	34	107	65	156	78	160
ppm @ 15% O_2 @ EF	34	103	65	149	78	153
lb/hr						
Carbon Monoxide Emissions						
ppm (dry)						
ppm @ 15% O_2	236	34	95	30	108	65
lb/hr						
Hydrocarbon Emissions						
ppm (dry)						
ppm @ 15% O_2	155	11	37	9	31	10
lb/hr						

Notes:

1. Air atomizing combustor used for smoke reduction.

TABLE 17
Facility 01
Summary of Results

Unit Type	Recuperated 2 Shaft R & D Gas Turbine				
Test Number	1	2	3	4	5
Test Date	6/74	7/74			7/74
% Rated Output (2.5 mw)(1)	100	100	100	100(2)	100
NO_x Controls	A-1	A-1	A-1	A-1	A-1
Ambient Conditions -					
Temp. °F	66	69			69
Pressure - psia	14.7	14.7			14.7
R.H. - %	76	59			59
Fuel	N.G.	N.G.	DF-2	Kerosene	NAPTHA
Stack Effluent					
Flow rate - lb/sec					
Temperature - °F					
Water vapor - Vol. %					
CO_2 - Vol. % (Actual)	2.13	1.97	2.6	2.5	2.45
O_2 - Vol. % (dry)	17.5	17.5	17.1	17.2	17.7
Visible Emissions - % opacity	0	0	0	0	0
Nitrogen Oxides Emissions					
ppm (dry)	93	69	196	186	173
ppm @ 15% O_2	159	119	302	294	315
ppm @ 15% O_2 @ EF	118	88	224	218	233
lb/hr					
Carbon Monoxide Emissions					
ppm (dry)					
ppm @ 15% O_2	17	39	19.5	32	20.4
lb/hr					
Hydrocarbon Emissions					
ppm (dry)					
ppm @ 15% O_2	3.1	4	4	4	2.0
lb/hr					

Notes:

1. Power output assumed same as for simple cycle version.
2. Assumed, based on fuel flow.

TABLE 18
Facility 01
Summary of Results

Unit Type: Recuperated 2 Shaft R&D Gas Turbine	
Test Number	1
Test Date	6/74
% Rated Output (2.5 mw)$^{(1)}$	0
NO_x Controls	A1
Ambient Temp. - °F	66
Pressure - PSIA	14.7
Relative Humidity - %	76
Fuel	N.G.
Stack Effluent	
Flow rate - lb/sec	--
Temperature - °F	--
Water vapor - Vol. %	--
CO_2 - Vol. % (dry)	0.99
O_2 - Vol. % (dry)	--
Visible Emissions - % opacity	
Nitrogen Oxides Emissions	
ppm (dry)	
ppm @ 15% O_2	31
ppm @ 15% O_2 @ EF	31
lb/hr	
Carbon Monoxide Emissions	
ppm (dry)	
ppm @ 15% O_2	4,695
lb/hr	
Hydrocarbon Emissions	
ppm (dry)	
ppm @ 15% O_2	6,011
lb/hr	

Notes: 1. Power output assumed same as for simple cycle version.

TABLE 19
Facility P1
Summary of Results

Unit Type : Production Gas Turbine

Test Number	1	2A	2B
Test Date			
% Rated Output (2.5 mw)	100	0	100
NO_x Controls	None		None*
Water-Fuel Ratio			
Fuel	N.G.		DF-1
Stack Effluent			
Flow rate - lb/sec	--	--	--
Temperature - ^oF			
Water vapor - Vol. %			
CO_2 - Vol. % (dry)			
O_2 - Vol. % (dry)	15.69	--	15.79
Visible Emissions - % opacity	1		7
Nitrogen Oxides Emissions			
ppm (dry)	88	--	109
ppm @ 15% O_2	99	66	125
" " " "$_2$ @ EF	94		118
lb/hr	16.7	--	20.7
Carbon Monoxide Emissions			
ppm (dry)	10	--	50
ppm @ 15% O_2	11	376	57
lb/hr	0.30	--	1.48
Hydrocarbon Emissions			
ppm (dry)			
ppm @ 15% O_2			
lb/hr			

*Combustor and fuel nozzle modified for smoke and particulate emissions control.

TABLE 20
Facility P2
Summary of Results

Unit Type : Production Gas Turbine

Test Number	1A	1B	2A	2B
Test Date	May 1972		May 1972	
% Rated Output (2.5 mw)	0	100	0	100
NO_x Controls	None		None (1)	
Water-Fuel Ratio				
Fuel	DF		DF	
Stack Effluent				
Flow rate - lb/sec	--	--	--	--
" " scfm	39,100	39,400	39,100	39,400
Temperature - ^oF	380	640	380	640
Water vapor - Vol. %				
CO_2 - Vol. % (dry)	1.10	2.26	1.00	2.53
O_2 - Vol. % (dry)	19.5	17.0	19.6	17.5
Visible Emissions - % opacity				
Nitrogen Oxides Emissions				
ppm (dry)	18	78	8.5	83
ppm @ 15% O_2	72	117	36.4	142
" " " " $_2$ @ EF(2)	72	111	36.4	134
lb/hr	--	23.6	--	26.2
Carbon Monoxide Emissions				
ppm (dry)	--	--	42	35
ppm @ 15% O_2	--	--	180	60
lb/hr				
Hydrocarbon Emissions				
ppm (dry)	8.0	4.0	7.0	4.0
ppm @ 15% O_2	32.0	6.0	30	6.9
lb/hr	--	--	--	--

Notes:
1. Engine Modification Made to Reduce Particulates
2. Design heat rate used for efficiency factor calculations.

TABLE 21
Facility Q & R
Summary of Results

Facility No.	Q						R	
Unit Type : TEST RIG	Production Combustor						R & D Combustor	
Test Number	1A	1B	2A	2B	2C	2D	1A	1B
Test Date								
% Rated Output (2.5 mw)	100		100				0	100
NO_x Controls	B-2		B-1				A-4 &	A-12
W/F Ratio	0	1.27	0	0.3	0.6	0.9		
Water-Fuel Ratio	DF-2		DF-1				DF-1	
Fuel								
Stack Effluent								
Flow rate - lb/sec								
Temperature - °F								
Water vapor - Vol. %								
CO_2 - Vol. % (dry)								
O_2 - Vol. % (dry)								
Visible Emissions - % opacity								
Nitrogen Oxides Emissions								
ppm (dry)								
ppm @ 15% O_2	113	68	154	83	44	32	140	50
" " " " @EF(1)	112	64	146	79	42	30	140	47
lb/hr								
Carbon Monoxide Emissions								
ppm (dry)								
ppm @ 15% O_2	45	54	33	30	36	38	200	4
lb/hr								
Hydrocarbon Emissions								
ppm (dry)								
ppm @ 15% O_2								
lb/hr								

Note: 1. Design heat rate of production engine used for efficiency calculations

TABLE 22
Facility S
Summary of Results

Unit Type :			
Test Number	1A	1B	2
Test Date			
% Rated Output (6.0 mw·base)	100	125	125
NO_x Controls	A-9		None
Water-Fuel Ratio			
Fuel	Distillate Oil		Natural Gas
Stack Effluent			
Flow rate - lb/sec	--	98,500	98,500
Temperature - ^oF			
Water vapor - Vol. %			
CO_2 - Vol. % (dry)			
O_2 - Vol. % (dry)	16.5	16.0	16.0
Visible Emissions - % opacity			
Nitrogen Oxides Emissions			
ppm (dry)	66	75	30
ppm @ 15% O_2	88	90	36
" " " "$_2$ @EF(1)	88	90	36
lb/hr		57	23
Carbon Monoxide Emissions			
ppm (dry)			
ppm @ 15% O_2			
lb/hr			
Hydrocarbon Emissions			
ppm (dry)			
ppm @ 15% O_2			
lb/hr			

Notes:

1. Efficiency factor of no benefit

TABLE 23
Facility T-1
Summary of Results

Unit Type : Pipeline Pumper

Test Number	1	2
Test Date		
% Rated Output (10.3 mw)	100	100
NO_x Controls	None	None
Water-Fuel Ratio		
Fuel	DF	N.G.
Stack Effluent		
Flow rate - lb/sec	125,400	125,400
Temperature - ^oF	775	775
Water vapor - Vol. %		
CO_2 - Vol. % (dry)		
O_2 - Vol. % (dry)	17.7	17.7
Visible Emissions - % opacity		
Nitrogen Oxides Emissions		
ppm (dry)	30	15
ppm @ 15% O_2	54	27
" " " "$_2$ @EF(1)	54	27
lb/hr	30	15
Carbon Monoxide Emissions		
ppm (dry)	90	50
ppm @ 15% O_2	162	90
lb/hr BTU	0.38	0.21
Hydrocarbon Emissions		
ppm (dry)		
ppm @ 15% O_2		
lb/hr		

Notes:
1. Efficiency factor of no benefit.

TABLE 24
Facility T2
Summary of Results

Unit Type : Production Pump Driver				
Test Number	1A	1B	2A	2B
Test Date	10/72		10/72	
% Rated Output (10.3 mw)	0	107	17	102
NO_x Controls	None		None	
Water-Fuel Ratio				
Fuel	DF-1		N.G.	
Stack Effluent				
Flow rate - lb/sec	30.3	167.1	87.2	161.1
Temperature - °F				
Water vapor - Vol. %				
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry)	18.0	17.6	18.5	17.7
Visible Emissions - % opacity				
Nitrogen Oxides Emissions				
ppm (dry)	13.4	85.4	12.1	40.8
ppm @ 15% O_2	26.8	151	5.04	74.2
" " " "$_2$ @ EF	26.8	146	5.04	71.5
lb/hr	2.68	79.35	6.51	36.2
Carbon Monoxide Emissions				
ppm (dry)	463	35.4	197	85.3
ppm @ 15% O_2	926	62.5	473	155
lb/hr	56.3	20.1	65.0	45.7
Hydrocarbon Emissions				
ppm ~~(dry)~~ (wet)	287	1.03	31.9	2.3
ppm @ 15% O_2	574	1.82	76.6	4.2
lb/hr	17.9	0.30	6.26	0.75

TABLE 25
Facility U1
Summary of Results

Unit Type: Field Test, Simple Cycle "Peaking" Power Plant

Test Number	1A		1B		1C		1D	
Test Date	5/72		5/72		5/72		5/72	
% Rated Output (15.1 peak MW)	86	100	86	100	86	100	86	100
NO_x Controls*[1]	B-1		B-1		B-1		B-1	
Water-Fuel Ratio	0.0		0.5		.75		1.0	
Fuel	← N.G. →							
Stack Effluent								
Flow rate - lb/sec								
Temperature - °F								
Water vapor - Vol. %								
CO_2 - Vol. % (dry)	1.8	--	1.8	--	1.8	--	1.8	--
O_2 - Vol. % (dry)(2)	17.6	17.3	17.6	17.3	17.6	17.3	17.6	17.3
Visible Emissions - % opacity								
Nitrogen Oxides Emissions								
ppm (dry)	68	82	34	44	24	40	15	29
ppm @ 15% O_2	120	149	60	80	42	73	26	53
" " " " w/EF(3)	120	148	60	79	42	72	26	53
lb/hr								
Carbon Monoxide Emissions								
ppm (dry)	28	46	109	87	137	277	214	233
ppm @ 15% O_2	49	84	192	158	242	504	378	424
lb/hr								
Hydrocarbon Emissions								
ppm (dry)	44	26	79	45	104	112	129	118
ppm @ 15% O_2	78	47	124	82	184	204	228	215
lb/hr								

Notes:
1. G.T. Equipped with smokeless combustor
2. Peak load O_2 percentage estimated
3. Efficiency factor of no benefit at base load.

TABLE 26

Facility U2

Summary of Results

Unit Type: Field Test, Simple Cycle "Peaking" Power Plant								
Test Number	1A	1B	2A	2B	3A	3B	4A	4B
Test Date			1/71 - 2/71					
% Rated Output (15.1 peak mw)	86	116	86	116	86	116	86	116
NO_x Controls	← B-1 →							
Water-Fuel Ratio	0.0	0.0	0.5	0.5	0.75	0.75	1.0	1.0
Fuel	← N.G. →							
Stack Effluent								
Flow rate - lb/sec								
Temperature - ^oF								
Water vapor - Vol. %								
CO_2 - Vol. % (dry)	1.8	--	1.8	--	1.8	--	1.8	--
O_2 - Vol. % (dry)(1)	17.6	17.3	17.6	17.3	17.6	17.3	17.6	17.3
Visible Emissions - % opacity								
Nitrogen Oxides Emissions								
ppm (dry)	59	66	22	26	16	19	9	15
ppm @ 15% O_2	104	120	39	47	28	35	16	27
" " " O_2 w/EF(2)	104	119	39	47	28	35	16	27
lb/hr								
Carbon Monoxide Emissions								
ppm (dry)	53	58	286	290	413	344	567	354
ppm @ 15% O_2	94	105	505	527	729	625	1001	644
lb/hr								
Hydrocarbon Emissions								
ppm (dry)	30	20	124	124	208	160	400	200
ppm @ 15% O_2	53	36	219	225	367	291	706	364
lb/hr								

Notes:

(1) Peak load O_2 percentage estimated.

(2) Efficiency factor of no benefit at base load.

TABLE 27

Facility U3

Summary of Results

Unit Type: Field Test, Simple Cycle Peaking Power Plant

Test Number	1	1
Test Date	8/69	8/69
% Rated Output (15.1 peak mw)	100	100
NO_x Controls	None	None
Water-Fuel Ratio	N/A	N/A
Fuel	NG	DF-2
Stack Effluent		
Flow rate - lb/sec	783,000	783,000
Temperature - °F	720	720
Water vapor - Vol. %	--	4.4
CO_2 - Vol. % (dry)	1.9	2.6
O_2 - Vol. % (dry)	17.6	17.4
Visible Emissions - % opacity		
Nitrogen Oxides Emissions		
ppm (dry)	35	70
ppm @ 15% O_2	62	124
" " " $"_2$ w/EF	62	123
lb/hr	--	--
Carbon Monoxide Emissions		
ppm (dry)	<10	<10
ppm @ 15% O_2	<18	<18
lb/hr		
Hydrocarbon Emissions		
ppm (dry)		
ppm @ 15% O_2		
lb/hr		

TABLE 28
Facility VI
Summary of Results

Unit Type : Simple Cycle "Peaking" Power Plant

Test Number (1)	1A	2A	1B	2B
Test Date	2/72	2/72	2/72	2/72
% Rated Output(17.2 peak mw)	110	106	110	106
NO_x Controls (5)	B-1	B-1	B-1	B-1
Water-Fuel Ratio	0	0	≈0.5 (4)	0.52
Fuel	N.G.	DF-2	N.G.	DF-2
Stack Effluent				
Flow rate - lb/sec	--	--	--	--
- DSCFM	--	213,300 (3)	--	--
Temperature - °F				
Water vapor - Vol. %	--	4.1 (3)	--	--
CO_2 - Vol. % (dry)	2.82	3.53	2.98	3.4
O_2 - Vol. % (dry)	15.3	15.1	15.1	14.9
Visible Emissions - % opacity				
Nitrogen Oxides Emissions				
ppm (dry)	88	110	38	54
ppm @ 15% O_2	92	113	38	53
" " " " w/EF (6)	92	113	38	53
lb/hr	--	--	--	--
Carbon Monoxide Emissions				
ppm (dry)	<10 (2)	<10	--	--
ppm @ 15% O_2	<10	<10	--	--
lb/hr	--	--	--	--
Hydrocarbon Emissions				
ppm (dry)	11.5 (2)	7.4 (2)	8.9	4.2
ppm @ 15% O_2	12.1	7.6	9.0	4.1
lb/hr	--	--	--	--

See Notes on attached sheet

Table 28
Facility VI

Notes:

1. Average of 3 runs
2. Data available for only 2 runs
3. " " " " 1 run.
4. Gas flow not recorded, but water flow comparable to Test 4.
5. Air atomization added to reduce plume opacity
6. Efficiency correction of no benefit.

Facility V1
Summary of Results

Unit Type: Field test, Simple Cycle Peaking Power Plant

Test Number	1C	2C
Test Date	2/72	2/72
% Rated Output (17.2 mw peak)	26.9	29.7
NO_x Controls(5)	None	None
Water-Fuel Ratio		
Fuel	N.G.	DF-2
Stack Effluent		
Flow Rate - lb/sec		
Temperature - °F		
Water vapor - Vol. %		
CO_2 - Vol. % (dry)	1.18	1.38
O_2 - Vol. % (dry)	18.1	17.78
Visible Emissions - % opacity		
Nitrogen Oxides Emissions		
ppm (dry)	22	31
ppm @ 15% O_2	45	55
ppm @ 15% O_2 w/EF(2)	45	55
lb/hr		
Carbon Monoxide Emissions		
ppm (dry)	150(3)	220
ppm @ 15% O_2	310	398
lb/hr		
Hydrocarbon Emissions		
ppm (dry)	20.3(3)	11.3(4)
ppm @ 15% O_2	42.0	20.4
lb/hr		

SEE NOTES ON ATTACHED SHEET

Table 29
Facility V1

Notes:
1. Average of 3 runs
2. " " 2 runs.
3. Data available for only 2 runs
4. " " " " 1 run.
5. Air Atomization added to reduce plume opacity
6. Efficiency correction of no benefit.

TABLE 30

Facility V2

Summary of Results

Unit Type : Field Test, Simple Cycle "Peaking" Power Plant

Test Number	1(1)	2A(2)	2B(3)	2C(3)
Test Date	1/73	1/73	1/73	1/73
% Rated Output(17.2 mw. Peak)	94.2	93.8	107.6	29.1
NO_x Controls(4)	B-1	B-1	B-1	B-1
Water-Fuel Ratio	0.43	0.43	0.42	0
Fuel	DF-2	DF-2	DF-2	DF-2
Stack Effluent				
Flow rate - lb/sec	--	--	--	--
Temperature - °F	972	972	1050	625
Water vapor - Vol. %				
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry)	15.5	15.2	14.4	17.5
Visible Emissions - % opacity	10	10	--	--
Nitrogen Oxides Emissions				
ppm (dry)	51.8	49.1	61	32
ppm @ 15% O_2	56.4	50.4	56	55
" " " "$_2$ w/EF(5)	56.4	50.4	56	55
lb/hr	59.3	53.4	--	--
Carbon Monoxide Emissions				
ppm (dry)	28.2	33.3	20	235
ppm @ 15% O_2	30.9	34.2	18.8	403
lb/hr				
Hydrocarbon Emissions				
ppm (dry)	0.77	0.50	2.5	9.5
ppm @ 15% O_2	0.84	0.52	2.2	16.3
lb/hr				

See notes on attached sheet.

Table 30
Facility V2

Notes:

1. Average of 3 runs, unit GT-2A, 49 point traverse
2. Average of 2 runs, unit GT-2B, " " "
3. Single point runs, " GT-2B, (one run each, no traverse)
4. Air atomization added to reduce plume opacity
5. Efficiency correction of no benefit.

TABLE 31
Facility V3
Summary of Results

Unit Type: Field Test, Simple Cycle Peaking Power Plant

Test Number (1)	1	2	3	4
Test Date				
% Rated Output (17.5 peak mw)	100	100	100	100
NO_x Controls	None	A	B-1	C-1
Water-Fuel Ratio	N/A	N/A	0.56	0.56
Fuel	DF-2	DF-2	DF-2	DF-2
Stack Effluent				
Flow rate - lb/sec				
Temperature - ^oF				
Water vapor - Vol. %				
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry)	15.5	15.5	15.5	15.5
Visible Emissions - % opacity				
Nitrogen Oxides Emissions				
ppm (dry)	150	100	75	57
ppm @ 15% O_2	164	109	82	62
" " " "$_2$ w/EF(2)	164	109	82	62
lb/hr	--	--	--	--
Carbon Monoxide Emissions				
ppm (dry)	--	--	500	--
ppm @ 15% O_2	--	--	547	--
lb/hr	--	--	--	--
Hydrocarbon Emissions				
ppm (dry)				
ppm @ 15% O_2				
lb/hr				

Notes:
(1) Number of tests used to provide this data base is unknown.
(2) Efficiency correction of no benefit.

TABLE 32
Facility V3
Summary of Results

Unit Type: Field test, simple cycle "peaking" power plant

Test Number(1)	5	6	7	8
Test Date				
% Rated Output (17.5 peak mw)	100	100	100	100
NO_x Controls	None	A	B-1	C-1
Water-Fuel Ratio	N/A	N/A	UNK	UNK
Fuel	NG	NG	NG	NG
Stack Effluent				
Flow rate - lb/sec				
Temperature - °F				
Water vapor - Vol. %				
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry)	15.5	15.5	15.5	15.5
Visible Emissions - % opacity				
Nitrogen Oxides Emissions				
ppm (dry)	85	68	39	30
ppm @ 15% O_2	93	74	43	33
" " " "$_2$ w/EF(2)	93	74	43	33
lb/hr	--	--	--	--
Carbon Monoxide Emissions				
ppm (dry)	500	--	100	--
ppm @ 15% O_2	547	--	110	--
lb/hr				
Hydrocarbon Emissions				
ppm (dry)				
ppm @ 15% O_2				
lb/hr				

Notes:
(1) No of tests used to provide data base is unknown.
(2) Efficiency correction of no benefit.

TABLE 33

Facility W

Summary of Results

Unit Type: Field test, Simple Cycle "Peaking" Power Plant

Test Number	1A(1)	1B(2)
Test Date	11/72	11/72
% Rated Output(21.3 mw max. peak)	23	85
NO_x Controls	D-1	D-1
Water-Fuel Ratio	0	0.6
Fuel	JP-5	JP-5
Stack Effluent-DSCFM	--	256,000
Flow rate - lb/sec	--	--
Temperature - °F	--	573
Water vapor - Vol. %		
CO_2 - Vol. % (dry)		
O_2 - Vol. % (dry)	18.5	17.8
Visible Emissions - % opacity	--	<10
Nitrogen Oxides Emissions		
ppm (dry)	20	31.1
ppm @ 15% O_2	48	62.2
" " " "$_2$ w/EF	48	56.7
lb/hr	--	61.2
Carbon Monoxide Emissions		
ppm (dry)	222	43.7
ppm @ 15% O_2	533	74
lb/hr		
Hydrocarbon Emissions		
ppm (dry)	--	2.9
ppm @ 15% O_2	--	5.4
lb/hr		

Notes: (1) 1 single point test (run 5)

(2) average of 3 runs of 44 traverse points each.

TABLE 34

Facility X

Summary of Results

Unit Type: Field Test, Simple Cycle "Peaking" Power Plant

Test Number	1A	1B	EST.(2)
Test Date	12/74	12/74	
% Rated Output(27.6 mw peak)	70	70	100
NO_x Controls		None	
Water-Fuel Ratio			
Fuel	DF-2	M(1)	M(1)
Stack Effluent			
Flow rate - lb/sec			
Temperature - ^oF			
Water vapor - Vol. %			
CO_2 - Vol. % (dry)			
O_2 - Vol. % (dry)	16.4	16.4	15.5
Visible Emissions - % opacity	--	Clear	--
- Von Brand	--	98-99	--
Nitrogen Oxides Emissions			
ppm (dry)	129	34	55
ppm @ 15% O_2	168	44	60
ppm @ 15% O_2, w/EF	141	37	50
lb/hr	--	--	--
Carbon Monoxide Emissions			
ppm (dry)	22	48	15
ppm @ 15% O_2	28	62	16
lb/hr			
Hydrocarbon Emissions			
ppm (dry)			
ppm @ 15% O_2			
lb/hr			

Notes: (1) Methanol fuel heat of combustion was 8550 Btu/lb.
(2) Fuel system sized for DF-2, which has about twice the heat of combustion, so full power output could not be achieved.

TABLE 35

Facility Y

Summary of Results

Unit Type: Fiel. Test, Peaking Unit

Test Number	1A	1B	2
Test Date	------12/71 - 2/72 ----------------		
% Rated Output(32.8 mw peak)	21	99	99
NO_x Controls	B-1	B-1	B-1
Water-Fuel Ratio	0	0	0.7
Fuel	--------------DF-2--------------------		
Stack Effluent			
Flow rate - lb/sec			
Temperature - °F			
Water vapor - Vol. %			
CO_2 - Vol. % (dry)	1.0	3.0	3.0
O_2 - Vol. % (dry)	17.0	14.1	14.1
Visible Emissions - % opacity	--	--	--
- Smoke spot (ASTM D-2156)	34	4.8	3.4
Nitrogen Oxides Emissions			
ppm (dry)	44	217	58
ppm @ 15% O_2	66	189	50
" " " " O_2 w/EF	66	163	43
lb/hr	--	--	--
Carbon Monoxide Emissions			
ppm (dry)	29	36	11
ppm @ 15% O_2	44	31	10
lb/hr			
Hydrocarbon Emissions			
ppm (dry)	≤ 0.96	≤ .96	0.80
ppm @ 15% O_2	≤ 1.4	≤ .84	0.70
lb/hr	--	--	--

TABLE 36
Facility Z1
Summary of Results

Unit Type: Combustor Rig Tests

Test Number	1A	1B	2A	2B
Test Date				
% Rated Output (1) (-- mw)	122	122	94	94
NO_x Controls	None	A-1	A-2 (2)	A-2 (3)
Water-Fuel Ratio				
Fuel	DF-2	DF-2	N.G.	N.G.
Stack Effluent				
Flow rate - lb/sec				
Temperature - ^oF				
Water vapor - Vol. %				
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry) (4)				
Visible Emissions - % opacity				
Nitrogen Oxides Emissions				
ppm (dry)	240	111	87	5
ppm @ 15% O_2 (4)				
" " " "$_2$ w/EF				
lb/hr				
Carbon Monoxide Emissions				
ppm (dry)				
ppm @ 15% O_2				
lb/hr				
Hydrocarbon Emissions				
ppm (dry)				
ppm @ 15% O_2				
lb/hr				

TABLE 36
Facility Z1

Notes:

(1) Power output not stated, full load assumed equivalent to a .017 fuel/air ratio.
(2) Primary airflow 35% of total airflow.
(3) Primary airflow 45% of total airflow.
(4) Percent O_2 in exhaust gas not noted in report.

TABLE 37
Facility Z2
Summary of Results

Unit Type : Combustor Rig Test, Peaking Unit

Test Number	1A/2A	1B	1C	1D	2B	2C
Test Date						
% Rated Output (--mw)(1)	100	100	100	100	100	100
NO_x Controls	None	A-3	A-1a&A-3	A-1b	B	A-1 & B
Water-Fuel Ratio					1.0	1.0
Fuel	-------------------------------DF-2 ------------------------					
Stack Effluent						
Flow rate - lb/sec						
Temperature - ^oF						
Water vapor - Vol. %						
CO_2 - Vol. % (dry)						
O_2 - Vol. % (dry)	14.0	14.0	14.0	14.0	14.0	14.0
Visible Emissions - % opacity						
Nitrogen Oxides Emissions						
ppm (dry)	202	140	115	96	40	19
ppm @ 15% O_2	173	120	99	82	34	16
" " " "$_2$ w/EF	149	104	85	71	29	14
lb/hr	--	--	--	--	--	--
Carbon Monoxide Emissions						
ppm (dry)						
ppm @ 15% O_2						
lb/hr						
Hydrocarbon Emissions						
ppm (dry)						
ppm @ 15% O_2						
lb/hr						

Notes: (1) output, not stated in report, is around 28 mw peak for engine.

TABLE 38
Facility AA1
Summary of Results

Unit Type: Scaled Combustor Rig Test

Test Number	1A	1B	2A	2B	3A	3B
Test Date						
% Rated Output (--(1) MW)(2)	100	100	100	100	100	100
NO_x Controls	None	A-5	None	A-5	None	D-2
Water-Fuel Ratio						7.0(4)
Fuel	N.G.	N.G.	DF-2	DF-2	DF-2	DF-2
Stack Effluent						
Flow rate - lb/sec	2	2	2	2	1.95	1.95
Temperature - °F						
Water vapor - Vol. %	4.83	4.83	4.83	4.83	--	--
CO_2 - Vol. % (dry)	4.83	5.55	5.55	5.55	--	--
O_2 - Vol. % (dry)	16.1	14.3	14.8	12.5	15.0(3)	15.0(3)
Visible Emissions - % opacity						
Nitrogen Oxides Emissions						
ppm (dry)	60	44	82	52	73	25
ppm @ 15% O_2	73	39	79	37	73	25
" " " " w/EF(5)	63	34	68	32	63	22
lb/hr	--	--	--	--	--	--
Carbon Monoxide Emissions						
ppm (dry)	43	43	70	30	50	200
ppm @ 15% O_2	53	39	68	21	50	200
lb/hr						
Hydrocarbon Emissions						
ppm (dry)	2	0	6.5	0	0	7.5
ppm @ 15% O_2	2.4	0	6.3	0	0	7.5
lb/hr						

SEE NOTES ON ATTACHED SHEET.

TABLE 38
Facility AA1

NOTES:

(1) Output not stated in report.
(2) 1800°F combustor exit temp. assumed equivalent to full load output.
(3) O_2 content assumed, with no change for steam injection.
(4) Fuel/air ratio assumed to be .017 at full load.
(5) Combustor efficiency assumed equiv. to full size unit.

TABLE 39
Facility AA2
Summary of Results

Unit Type : Scaled Combustor Rig Test

Test Number	1A	1B	1C	2A	2B
Test Date					
% Rated Output((1) MW)(4)	100	100	100	100	100
NO_x Controls	None	A-1a	A-1b	None	A-1b
Water-Fuel Ratio					
Fuel	N.G.	N.G.	N.G.	DF-2	DF-2
Stack Effluent					
Flow rate - lb/sec	1.95				
Temperature - °F					
Water vapor - Vol. %					
CO_2 - Vol. % (dry)					
O_2 - Vol. % (dry)(2)	16.1	16.1	16.1	14.8	14.8
Visible Emissions - % opacity		--	--	--	--
- ASTM smoke		0	0	41	26
Nitrogen Oxides Emissions					
ppm (dry)	61	53	49	70	61
ppm @ 15% O_2	75	65	60	68	59
" " " O_2 w/EF(3)	65	56	52	59	51
lb/hr					
Carbon Monoxide Emissions					
ppm (dry)	38	26	21	105	85
ppm @ 15% O_2	47	32	26	102	82
lb/hr					
Hydrocarbon Emissions					
ppm (dry)	7	--	5	6	6
ppm @ 15% O_2	9	--	6	6	6
lb/hr					

Notes: (1) output not stated in report. (2) Percent O_2 not stated in report, assumed same as for unit with EGR with no change for dry controls. (3) Combustor efficiency assumed equivalent to full size combustor.
(4) 1800°F output temp. assumed equivalent to full size load output.

Facility BA 1

Summary of Results

Unit Type	Scaled Combustor Rig Test(1)						
Test Number	1A	1B	1C	1D	2A	2B	2C
Test Date	5/73				4/74		
% Rated Output (--MW)		--	--		--	--	
NO_x Controls		None		E	A-2	E	
Water-Fuel Ratio							
Fuel	DF-2	SCG(2)	DF-2	SCG(2)	DF-2	DF-2	SCG(2)
Stack Effluent							
Temperature - F	1800	1742	2082	2112	1950	2173	2220
O_2 - Vol % (Dry)	--	--	--	--	--	--	--
Inlet Temperature F	620-770						
Inlet Pressure - Atm.	3-6						
Fuel/Air Ratio	.026 - .028						
Space Velocity	--	--	--	--	--	--	--
Reference velocity FT/SEC	130	92	73	105	74	112	70
Combustor Pressure PSIA	42	51.1	56.5	44.0	47.2	43.9	46.7
Nitrogen Oxides Emissions							
ppm (dry)							
ppm @ 15% O_2	60	9	<1	<1	6	5	1
lb/hr							
Carbon Monoxide Emissions							
ppm (dry)	18	160	10	40	20	30	26
ppm @ 15% O_2							
lb/hr							
Hydrocarbon Emissions							
ppm (dry)	8	80	--	14	18	1	1
ppm @ 15% O_2							
lb/hr							

Notes: See attached sheet

TABLE 40
Facility BA 1

Notes:

(1) Same test rig used for all tests, with 6"D combustors installed for tests 1A, 1B and 2A and 6"D catalyst cores installed for tests 1C, 1D, 2B and 2C

(2) molar composition of synthetic coal gas was = H_2 - 15.4%, CO = 11.6% methane - 5.1%, CO_2 = 10.1%, N_2 - 57.8%. Lower heating value = 126 Btu/ft.^{3}.

Facility BA 2

Summary of Results

Unit Type: Scaled Combustor Rig Test(1)

Test Number	1A	1B	1C	1D	1E	1F	1G	1H
Test Date								
% Rated Output								
NO_x Controls	E							
Water-Fuel Ratio								
Fuel(2)	M	P	DF-2	JP-4	Jet-A	SCG	Doped P	Doped DF-2
Inlet Temp. - °C	550	365	380	405	400	220	250	360
Fuel/Air Ratio	.024	.022	.027	.022	.027	.313	.032	.025
Space Velocity (1000 hr.$^{-1}$)	120	100	130	180	140	300	190	130
REF. Velocity - Ft/Sec.	---	---	---	---	---	---	---	---
Pressure - ATM	1							
Visible Emissions - % opacity								
Nitrogen Oxides Emissions								
ppm (dry)	1	1	4	1	1	1	80(3)	450(3)
ppm @ 15% O_2								
ppm @ 15% O_2 w/EF								
lb/hr								
Carbon Monoxide Emissions								
ppm (dry)	48	10	48	15	32	25	10	8
ppm @ 15% O_2								
lb/hr								
Hydrocarbon Emissions								
ppm (dry)	10	3	0	0	6	2	5	3
ppm @ 15% O_2								
lb/hr								

NOTES: See attached sheet

TABLE 41

Facility BA 2

NOTES:

(1) 1" nominal diameter test rig used for all tests.

(2) M = Methane, P = Propane, Doped P is propane with 0.17% N_2 as NH, Doped DF-2 contains 0.94% N_2 as pyridine, SCG = Synthetic coal gas.

(3) Conversion of fuel N_2 is 70-90% compared to about 50% for a diffusion flame burner.

TABLE 42
Facility CA
Summary of Results

Unit Type : Aircraft Engine Test Rig(1)

Test Number	1A	1B	1C	1D
Test Date				
% Rated Output (--(2) mw)	100	100	100	100
NO_x Controls	None	A-13	A-2	A-14
Water-Fuel Ratio				
Fuel	Jet - A			
Stack Effluent				
Flow rate - lb/sec				
Temperature - ^oF				
Water vapor - Vol. %				
CO_2 - Vol. % (dry)				
O_2 - Vol. % (dry)	13.6	13.6	13.6	13.6
Visible Emissions - % opacity	-	-	-	-
- SAE Smoke #	10	1	6	14
Nitrogen Oxides Emissions				
ppm (dry)	440	190	288	173
ppm @ 15% O_2	357	154	234	140
" " " O_2 w/EF(3)	248	107	162	97
lb/hr	--	--	--	--
Carbon Monoxide Emissions				
ppm (dry)				
ppm @ 15% O_2				
lb/hr				
Hydrocarbon Emissions				
ppm (dry)				
ppm @ 15% O_2				
lb/hr				

TES: See attached sheet

TABLE 42

Facility CA

Notes:

(1) Test rig simulates engine combustor inlet conditions except for pressure (6.8 atmospheres versus 21.7 produced in engine). Correction factors applied to compensate for this difference.

(2) Engine produces 44,300 lb thrust at takeoff (about 30 mw @ 1.1/2)

(3) Based on SFC of .349 #Fuel/#thrust and an estimated 1.1 lb thrust per horsepower.

TABLE 43
Facility DA
Summary of Results

Unit Type: Advanced Aircraft Engine Combustor Test Rig(1)							
Test Number	1	2A	2B	3A	3B	4A	4B
Test Date							
% Rated Output ($\overset{(3)}{--}$ mw)	100	69	100	69	100	69	100
NO_x Controls	None	None		A-4		A-5	
Water-Fuel Ratio							
Fuel	----------Jet-A----------						
Stack Effluent							
Flow Rate - lb/sec							
Temperature - ^oF							
Water vapor - Vol. %							
CO_2 - Vol. % (dry)							
O_2 - Vol. % (dry)	13	14.5	13	14.5	13	14.5	13
Visible Emissions - SAE Smoke #	11.4	12		< 15		< 15	
Nitrogen Oxides Emissions(2)							
ppm (dry)	529	565	620	167	227	244	317
ppm @ 15% O_2	397	522	465	154	170	225	238
ppm @ 15% O_2 w/ EF	260	522	304	154	111	225	158
lb/hr							

Notes:

(1) Test rig data extrapolated to engine conditions for current production engines.
(2) Fuel/air ratio for most industrial gas turbines at base load is .020. At this F/A, power output is about 69% and efficiency about 48% of that at take off.
(3) Engine is in 50,000 lb thrust class, with 100% load being takeoff mode. (about 34.4 mw)

TABLE 44

Facility EA

Summary of Results

Test Type:	Field Test					
Unit Type:	Production Simple Cycle					
Test Number:	1A	1B	2A	2B	3A	3B
Test Date:	7/73					
% Rated Output (26 MW base)	20	85	20	85	20	85
NO_x Controls	None		None		None	
Water-Fuel Ratio						
Fuel	N.G.		DF-2(1)		Heavy Dist.(2)	
Stack Effluent						
Flow rate - lb/sec						
Temperature - °F		850		850		850
Water vapor - Vol. %						
CO_2 - Vol. % (dry)						
O_2 - Vol. % (dry)	18.8	16.8	18.7	16.6	18.6	17.2
Visible Emissions - Smoke SBT (0-2156)	0	0	4.7	5.5	4.2	5.1
Nitrogen Oxides Emissions						
ppm (dry)	52	72	48	137	28	127
ppm @ 15% O_2	142	103	125	187	70	201
ppm @ 15% O_2 @ E.F.		89		161		173
lb/hr						
Carbon Monoxide Emissions						
ppm (dry)	20	3	15	6	24	7.5
ppm @ 15% O_2	55	4.3	39	8.2	60	11.8
lb/hr						
Hydrocarbon Emissions						
ppm (dry)	0.65	0.25	0.5	0.5	0.56	0.25
ppm @ 15% O_2	1.8	0.36	1.3	0.68	1.4	0.39
lb/hr						

NOTES:

(1) Sulfur content of DF-2 ranged from 0.024 to 0.0255 percent and averaged 0.136 percen

(2) Sulfur content of heavy distillate ranged from 0.17 to 0.24 percent and averaged 0.2 percent.

TABLE 45

Facility FA

Summary of Results(1)

Test type - Field			
Unit Type - Production simple cycle			
Test Number	1A	1B	1C
Test Date	1/74 - 8/74		
% Rated Output (51.7 mw base)	0	100	100
NO_x Controls	B 1		
Water-Fuel Ratio	0	0	1.12
Fuel	DF-2		
Stack Effluent			
Flow rate - lb/sec	--	600	--
Temperature - °F	--	900	--
Water vapor - Vol. %			
CO_2 - Vol. % (dry)			
O_2 - Vol. % (dry)	19.68	16.62	16.62
Visible Emissions - % opacity	12	8	--
Nitrogen Oxides Emissions			
ppm (dry)	38	230	43
ppm @ 15% O_2	172	315	58
" " " " w/EF	172	274	55
lb/hr	--	809	--
Carbon Monoxide Emissions			
ppm (dry)	55	22	80
ppm @ 15% O_2	250	30	110
lb/hr	--	47.2	--
Hydrocarbon Emissions			
ppm (dry)	--	--	--
ppm @ 15% O_2			
lb/hr			

Note: 1) Data corrected to 150 conditions.

TABLE 46
Facility GA
Summary of Results

Test type - Field

Unit Type - Production Simple Cycle

Test Number	1	2
Test Date	12/72	10/72 and 12/72
% Rated Output (52.9 mw)(1)	100	100
NO_x Controls	None	
Water-Fuel Ratio	--	--
Fuel	DF-2	Crude
Stack Effluent		
Flow rate - lb/sec		
Temperature - °F		
Water vapor - Vol. %		
CO_2 - Vol. % (dry)		
O_2 - Vol. % (dry)	16.0	16.0
Visible Emissions - % opacity		
Nitrogen Oxides Emissions		
ppm (dry)	128	140
ppm @ 15% O_2	154	168
" " " " w/EF	135	147
lb/hr	--	--
Carbon Monoxide Emissions		
ppm (dry)		
ppm @ 15% O_2		
lb/hr		
Hydrocarbon Emissions		
ppm (dry)		
ppm @ 15% O_2		
lb/hr		

Note: (1) A firing temp. of 1730°F was assumed equivalent to full load.

TABLE 47

Facility HA 1
Summary of Results

Test Type - Field

Unit Type - Production Simple Cycle

Test Number	1A	1B	1C	1D	1E	1F
Test Date						
% Rated Output (60.4 mw base)	98.3	99.5	101.1	102.3	102.8	103.5
NO_x Controls	B-1					
Water-Fuel Ratio	0	0.23	0.45	0.66	0.87	1.07
Fuel	DF-2					
Stack Effluent						
Flow rate - lb/sec	512	512	512	511	508	506
Temperature - ^oF						
Water vapor - Vol. %						
CO_2 - Vol. % (dry)						
O_2 - Vol. % (dry)	15.60	15.50	15.40	15.30	15.20	15.10
Visible Emissions - % opacity						
Nitrogen Oxides Emissions						
ppm (dry)	147	98.8	67.4	44.7	32.5	25.5
ppm @ 15% O_2	163	108	72.3	47.1	33.6	25.9
ppm @ 15% O_2 w/ EF	130	85.8	57.4	37.4	26.7	20.6
lb/hr	---	---	---	---	---	---
Carbon Monoxide Emissions						
ppm (dry)	0	0	0	0	4.0	9.0
ppm @ 15% O_2	0	0	0	0	4.1	9.2
lb/hr	0	0	0	0	---	---
Hydrocarbon Emissions						
ppm (dry)	0	0	0	0	0	0
ppm @ 15% O_2	0	0	0	0	0	0
lb/hr	0	0	0	0	0	0

TABLE 48

Facility HA 2

Summary of Results(1)

Test type - Field

Unit Type - Production Simple Cycle

Test Number	1A(2)	1B	1C	1D	1E
Test Date					
% Rated Output (61.5 mw base)	82.9	84.6	86.2	87.8	89.4
NO_x Controls	B 1				
Water-Fuel Ratio	0	0.27	0.52	0.73	0.98
Fuel	NG				
Stack Effluent					
Flow rate - lb/sec	442	451	453	452	452
Temperature - ^oF					
Water vapor - Vol. %					
CO_2 - Vol. % (dry)					
O_2 - Vol. % (dry)	15.67	15.40	15.25	15.10	14.90
Visible Emissions - % opacity					
Nitrogen Oxides Emissions					
ppm (dry)	99.0	60.7	37.4	24.6	13.4
ppm @ 15% O_2	111	65.0	39.0	25.0	13.2
" " " " w/ EF	87.8	51.2	30.7	19.7	10.4
lb/hr	--	--	--	--	--
Carbon Monoxide Emissions					
ppm (dry)	2.14	3.00	3.00	3.00	10.00
ppm @ 15% O_2	2.41	3.21	3.13	3.05	9.84
lb/hr	--	--	--	--	--
Hydrocarbon Emissions					
ppm (dry)	2.44	1.06	1.06	1.07	2.15
ppm @ 15% O_2	2.75	1.14	1.11	1.09	2.11
lb/hr	--	--	--	--	--

Notes: (1) Gas turbine inlet air at 100% relative humidity (air saturated using evaporative coolers).

(2) Average of 7 tests.

TABLE 49

Facility J A

Summary of Results

Test type: Field

Unit Type : Production Regenerative Cycle

Test Number	1(1)	2(2)
Test Date	--	--
% Rated Output (58.6 mw base)	82.4	90.4
NO_x Controls	None	
Water-Fuel Ratio		
Fuel	DF-2	
Stack Effluent		
Flow rate - lb/sec		
Temperature - ^oF		
Water vapor - Vol. %	4.52	
CO_2 - Vol. % (dry)		
O_2 - Vol. % (dry)	16.45	16.15
Visible Emissions - % opacity		
Nitrogen Oxides Emissions		
ppm (dry)	232	295
ppm @ 15% O_2	306	366
" " " w/EF	199	238
lb/hr	--	--
Carbon Monoxide Emissions		
ppm (dry)	4.69	4.01
ppm @ 15% O_2	6.18	4.96
lb/hr	--	--
Hydrocarbon Emissions		
ppm (dry)		
ppm @ 15% O_2		
lb/hr		

Notes: (1) Average of 3 tests
(2) Average of 2 tests

REFERENCES FOR APPENDIX C

1. EPA test report GT-8747-R, Rev.1, dated January 25, 1972, entitled, "Exhaust Emission Test, Airesearch Aircraft Propulsion and Auxiliary Power Gas Turbine Engines".
2. Letter dated February 13, 1976 from D. G. Medigovich, Airesearch Manufacturing Company of Arizona, to Don R. Goodwin, EPA, and accompanying status report dated January 30, 1976.
3. Letter dated January 22, 1976, and coded ES:JMH:0401:012276, from J. M. Haasis, Airesearch, to E. A. Noble, EPA.
4. Letter dated August 26, 1974, from R. Kress, Solar, to Don R. Goodwin, EPA.
5. Report, "General Motors Response to Preliminary (Draft) Proposed Standards for Control of Air Pollution from Stationary Gas Turbines", dated March 21, 1973, and corrected March 28, 1973.
6. Letter dated October 17, 1972, from W. P. Slichter, Bell Laboratories to Don R. Goodwin, EPA, and accompanying test report.
7. Sawyer's Gas Turbine Catalog, 1974 edition.
8. Letter dated April 23, 1976 from E. A. Noble, EPA, to R. Kress, Solar, documenting information provided verbally by A. Finklestein on March 26, 1976.
9. Letter dated November 10, 1975 from O. M. Sievert, Solar, to Don R. Goodwin, EPA, and accompanying submittal.
10. NATCO Engineering Data Sheet 799-052-003, dated December 20, 1971, on the Viking KG2-3 Exhaust Emission Analysis (Diesel #2 fuel).
11. Record of telephone conversation with Bill Wittner of North American Turbine Company on September 12, 1972.

12. Record of telephone conversation with S. Lombardo of Curtis-Wright on May 4, 1972.
13. Record of telephone conversation with S. Lombardo of Curtis-Wright on September 7, 1972.
14. Letter dated April 18, 1972, from R. W. Wheeler, Alyeska Pipeline Service Company, to T. Kittleman, EPA, and accompanying data.
15. Letter dated May 11, 1972, from R. M. Cummings, Alyeska Pipeline Service Company, to J. A. Eddinger, EPA, and accompanying data.
16. Test Report GTT-21 from Cooper-Bessemer, entitled, "Exhaust Gas Emission Test, RT-125 Gas Turbine Package MO-280RP, Alyeska Pipeline Service Company."
17. Test Report No. 11 from the San Diego Gas and Electric Company entitled, "Report on the Sampling and Analysis of Emissions from Kearny Mesa Turbine GT-2B."
18. EPA Test Report 73-TRB-2, "San Diego Gas and Electric Company Kearny Mesa Gas Turbine, San Diego, California", dated March, 1973.
19. Letter dated October 16, 1972, from O. J. Ortega, Southern California Edison Company, to Don R. Goodwin, EPA.
20. Letter dated November 10, 1972, from O. J. Ortega, Southern California Edison Company, to Don R. Goodwin, EPA.
21. ASME Publication 72-GT-53, "Nitric Oxide Abatement in Heavy Duty Gas Turbine Combustors by Means of Aerodynamics and Water Injection", by M. B. Hilt and R. H. Johnson, General Electric Company.
22. ASME Publication 72-JPG-GT-2, "Recent Field Tests for Control of Exhaust Emissions from a 35-MW Gas Turbine", by M. J. Ambrose and E. S. Obidinski, Westinghouse Electric Company.
23. Appendix 3.a 1-2 of letter dated January 8, 1976, from S. M. DeCorso, Westinghouse Electric Corporation, to Don R. Goodwin, EPA.

24. ASME Publication 72-GT-22, "Formation and Control of Oxides of Nitrogen Emissions from Gas Turbine Combustor Systems", P. P. Singh, W. E. Young, and M. J. Ambrose, Westinghouse Electric Company.

25. Appendix 3.a 1-16 of letter dated January 8, 1976, from S. M. DeCorso, Westinghouse Electric Corporation, to Don R. Goodwin, EPA.

26. Report from Engelhard Industries Division of Engelhard Minerals and Chemical Corporation, entitled, "Catalytically-Supported Thermal Combustion for Emission Control", no date.

27. NASA Report CR-134736, "Experimental Clean Combustor Program, Phase 1, Final Report", by the Pratt & Whitney Division of United Technologies Corporation and dated October, 1975.

28. EPA test report 73-TRB-1, "San Diego Gas & Electric Company South Bay Gas Turbine, San Diego, California", dated March, 1973.

29. ASME Publication 75-PWR-22, "Gas Turbine Emissions and Performance on Methanol Fuel," by R. D. Klapatch, TP&M.

30. MASA report NASACR 134737, "Experimental Clean Combustor Program, Phase 1 Final Report", by the General Electric Company and undated.

31. Letter dated January 9, 1976, from R. H. Gaylord, Turbodyne Corporation, to Don R. Goodwin, EPA, and enclosures.

32. Record of telephone call on February 5, 1976, from E. A. Noble, EPA, to H. Gaylord, Turbodyne division of Worthington Turbine International.

33. Handout on emissions from an FPC unit at Bartow, Florida, provided by E. W. Zeltmann, General Electric, at a meeting with EPA personnel on August 19, 1975.

34. Letter dated October 31, 1975, from E. Zeltmann, General Electric Company, to K. R. Durkee, EPA, and enclosures.

35. ASME Publication 75-GT-68, "Exhaust Emissions from a 25-MW Gas Turbine Firing Heavy and Light Distillate Fuel Oils and Natural Gas", D. E. Carl, E. S. Obidinski and C. A. Jersey, Westinghouse Electric Corporation.

APPENDIX D

Emission Measurement and Monitoring

D.1 Emission Measurement Methods

The new source performance standard for gas turbines was based upon the results of tests conducted by gas turbine users and manufacturers and by EPA. A careful review of the available gas turbine user and manufacturer test reports showed that the sampling and instrumental analysis procedures used were, generally speaking, based on the gas turbine test methods most frequently recommended in the literature, i.e., the EPA Mobile Sources Jet Engine Test Method[1] and the SAE Aerospace Recommended Practice (ARP1256)[2]. For this reason, the test results were accepted as reliable and were used as part of the data base for the standard.

The EPA tests were done using a working draft of an instrumental method for stationary gas turbines. Information from the two literature methods cited above were used in the development of this method. The basic requirements of the instrumental method are: (1) performance specifications for the measurement system; (2) a sampling traverse to canvass emissions from the stack; (3) an O_2 correction to adjust for variations in excess air.

In all, EPA conducted three gas turbine NSPS tests; in each instance, the instrumental method was compared against the manual EPA methods for NO_x and O_2 (i.e., Methods 7 and 3, respectively). The instrumental analyzers used in the tests were calibrated with vendor-certified gas mixtures, which were reanalyzed by the appropriate wet chemical methods. Although zero and calibration drift data for the analyzers were not reported, the calibration of the instruments was checked frequently during the test program. Sulfur dioxide emissions were not actually measured during the test series; rather, the sulfur content of the fuel was determined by analysis, and the theoretical SO_2 level was calculated. The calculation procedure is considered to be at least as accurate as actual SO_2 measurement.

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[1] Federal Register, July 17, 1973, Vol. 38, #136, part 2, p. 19088-19103. Control of Air Pollution from Aircraft and Aircraft Engines. Emission Standards and Test Procedures for Aircraft.

[2] Society of Automotive Engineers. Procedure for the Continuous Sampling and Measurement of Gaseous Emissions from Aircraft Turbine Engines. (ARP1256) issued 10-1-71.

The results of the EPA tests showed that the precision of the instrumental data was better than Method 7. Precision was estimated from the standard deviation of the data sets. In two of the EPA tests the instrumental data agreed to within 3 ppm of the Method 7 data. However, the standard deviation for instrumental data was less than 2 ppm and for Method 7 approximately 11 ppm.

After the test program, Method 20 - "Determination of Nitrogen Oxide, Sulfur Dioxide, and Oxygen Emissions from Stationary Gas Turbines," was developed from the working draft. Quantitative values for the Method 20 performance specifications were established during the test program using procedures described in the continuous monitoring regulations of 40CFR60. Recently, EPA validated the procedures of Method 20, by conducting a simulated performance test on a controlled gas turbine.

D.2 Continuous Monitoring

Although EPA has established performance specifications for continuous monitoring instruments in Appendix B of 40 CFR part 60, the specifications were developed with the understanding that the instruments would be used to monitor large industrial sources which operate continuously and have well developed velocity and temperature profiles. Gas turbines do not fit into this source category. Furthermore, EPA has not, to date, conducted any tests to determine performance specifications for monitoring this type of source. Therefore, EPA cannot recommend continuous monitoring instruments for gas turbines.

Additional factors which could affect the feasibility of continuous monitoring of gas turbine emissions are:

a. Gas stratification found in some turbine exhaust stream makes location of the sampling point a critical factor.

b. There is a lack of personnel at remote turbines to routinely check and maintain the sampling equipment.

c. High costs are associated with emission monitoring at turbine facilities because a single site may have multiple turbines. For example, the costs for opacity monitoring have been estimated $20,000 capital and $8,500 annual operating. Since an opacity monitor is an in-stack type monitor, each turbine would require a separate system; it would not be possible to time-share its use at adjacent turbines, as it would be with an extractive type unit. The costs of gaseous pollutant monitors for NO_x and O_2 are estimated at $30,000 capital (per monitor) and $16,000 annual operating. Facilities which have more than one turbine could reduce this total cost by selecting an extractive type system and by time-sharing the system between several turbines.

D.3 Performance Test Method

Method 20 - "Determination of Nitrogen Oxide, Sulfur Dioxide, and Oxygen Emissions from Stationary Gas Turbines," is recommended as the performance test method. The use of the Mobile Source[1] or SAE[2] test procedures was considered but was rejected because these methods specify the use of particular types of instruments. However, both the SAE and Mobile Source test methods are acceptable alternative methods, if the selected instrument models meet the performance specifications of Method 20.

The selection of Method 20 for performance testing was based on the results of EPA gas turbine field tests (Section DJ). Method 20 includes the following: (1) measurement system design criteria; (2) measurement system performance specifications (including analyzer span drift, zero drift, linearity check, response time and interference checks) and performance test procedures; and (3) procedures for emission sampling.

EPA Method 6 is recommended as the performance test method for SO_2. The sample point location for Method 6 is prescribed in Method 20. The sample volume of Method 6 is not fixed and may be increased if the expected SO_2 concentration is very small. In lieu of measuring SO_2 the standard permits the measurement of the sulfur content of the fuel by the applicable ASTM method. The sulfur content is then used to calculate the SO_2 concentration.

The cost of conducting a Method 20 emission test is estimated at $4000 to $6000 per turbine. The testing costs per turbine may be lower if several turbines at a single site are to be tested.

APPENDIX E - AIR QUALITY ANALYSES

Dispersion analyses have been conducted for nine individual units and for several cluster arrangements as are discussed in Section 6.1. Concentrations are estimated at distances of 0.1, 0.2, 0.5, 2.0 and 20 kilometers from the downwind edge of the units. Because of the large volume of data, the results are reported for only one set of emission levels. For the analyses of individual turbines and the cluster arrangements having all turbines operating in the same mode, concentration estimates for other emission rates change proportionately.

Table E-1

GARRETT AIRESEARCH

GTC 8S-90; JET A FUEL
(GPUT-90)

Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance (Km)	3-Hour SO_2 Average 0.8% S Q = .38 g/sec ($\mu g/m^3$)	24-Hour SO_2 Average 0.8% S Q = .38 g/sec ($\mu g/m^3$)	Annual SO_2 Average 0.8% S Q = .38 g/sec ($\mu g/m^3$)
.1	134.9	94.4	9.3
.2	50.4	34.4	3.3
.5	22.6	15.5	1.8
2.0	14.2	3.0	0.4
20.0	2.1	0.3	—

	1-Hour CO Average 321 ppm Q = .4 g/sec. (mg/m^3)	8-Hour CO Average 321 ppm Q = .4 g/sec. (mg/m^3)	Annual NO_x Average 125 ppm Q = .26 g/sec. ($\mu g/m^3$)
.1	0.2	0.1	6.4
.2	0.1	—	2.3
.5	—	—	1.3
2.0	—	—	0.2
20.0	—	—	—

— Concentrations less than 0.1

Table E-2

GARRETT AIRESEARCH

GTC 8S-90;JET A FUEL
(GPUT-90)
Maximum Ground Level Concentration; Dallas, Texas, 1964
20 Unit Cluster (see Figure 6-1)

Downwind Distance (km)	1-Hour CO Average 321 ppm Q = .4 g/sec. (mg/m^3)	8-Hour CO Average 321 ppm Q = .4 g/sec. (mg/m^3)	3-Hour SO_2 Average .8% S Q = .38 g/sec. (μg/m^3)	24-Hour SO_2 Average .8% S Q = .38 g/sec. (μg/m^3)
.1	0.3	0.3	267.0	216.8
.2	0.1	0.2	167.9	160.5
.5	0.1	0.1	126.9	133.6
2.0	0.1	—	41.8	36.3
20.0	—	—	1.7	1.2

— Concentrations less than 0.1

Table E-3

SOLAR

SATURN; KEROSENE FUEL

Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance (km)	3-Hour SO_2 Average 0.8% S Q = 1.33 g/sec. ($\mu g/m^3$)	24-Hour SO_2 Average 0.8% S Q = 1.33 g/sec. ($\mu g/m^3$)	Annual SO_2 Average 0.8% S Q = 1.33 g/sec. ($\mu g/m^3$)
.1	158.8	35.9	1.4
.2	71.1	23.0	1.9
.5	32.9	21.3	2.0
2.0	16.9	5.5	0.8
20.0	3.7	4.8	—

	1-Hour CO Average 250 ppm Q = 1.12 g/sec (mg/m^3)	8-Hour CO Average 250 ppm Q = 1.12 g/sec (mg/m^3)	Annual NO_x Average 125 ppm Q = .92 g/sec ($\mu g/m^3$)
.1	0.2	0.1	0.9
.2	0.1	0.1	1.3
.5	—	—	1.4
2.0	—	—	0.6
20.0	—	—	—

— Concentrations less than 0.1

Table _-4

SOLAR

SATURN; KEROSENE FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964

5 and 22 Unit Cluster (see Figure 6-1)

Downwind Distance	1-Hour CO Average	8-Hour CO Average	3-Hour SO_2 Average		24-Hour SO_2 Average	
	163 ppm Q = .73 g/sec.	163 ppm Q = .73 g/sec.	.8% S Q=1.33g/sec	.3% S Q=.5g/sec.	.8% S Q=1.33g/sec	.3% S Q=.5g/sec.
(km)	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
	22 Unit Cluster					
.1	1.6	0.9	2655.0	998.0	692.0	260.0
.2	0.9	0.6	1391.0	523.0	482.0	181.0
.5	0.4	0.3	537.3	202.0	260.0	98.0
2.0	0.1	0.1	82.7	31.1	48.8	18.3
20.0	—	—	2.4	0.9	1.4	0.5
	5 Unit Cluster					
.1	0.3	0.2	627.0	236.0	157.5	59.2
.2	0.2	0.1	349.0	131.0	123.6	46.5
.5	0.1	0.1	119.0	44.7	62.8	23.6
2.0	—	—	19.2	7.2	11.4	4.3
20.0	—	—	0.5	0.2	0.3	0.1

— Concentrations less than 0.1

Table E-5

SOLAR

CENTAUR; KEROSENE FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance (km)	3-Hour SO_2 Average 0.8% S Q = 3.58 g/sec. ($\mu g/m^3$)	24-Hour SO_2 Average 0.8% S Q = 3.58 g/sec. ($\mu g/m^3$)	Annual SO_2 Average 0.8% S Q = 3.58 g/sec ($\mu g/m^3$)
.1	135.9	22.9	0.7
.2	93.6	24.4	1.4
.5	34.5	13.1	1.1
2.0	26.8	10.7	1.4
20.0	7.8	1.2	0.9

	1-Hour CO Average 250 ppm Q = 3.05 g/sec. (mg/m^3)	8-Hour CO Average 250 ppm Q = 3.05 g/sec. (mg/m^3)	Annual NO_x Average 125 ppm Q = 2.5 g/sec. ($\mu g/m^3$)
.1	0.2	0.1	0.5
.2	0.1	0.1	0.9
.5	—	—	0.7
2.0	—	—	0.9
20.0	—	—	0.6

— Concentrations less than 0.1

Table 6

SOLAR
CENTAUR; ~~KEROSENE~~ FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964
5 and 22 Unit Cluster (see Figure 6-1)

Downwind Distance (km)	1-Hour CO Average 250 ppm Q=3.05 g/sec (mg/m^3)	8-Hour CO Average 250 ppm Q=3.05 g/sec (mg/m^3)	3-Hour SO_2 Average .8% S Q=3.58 g/sec ($\mu g/m^3$)	3-Hour SO_2 Average .3% S Q=1.34 g/sec ($\mu g/m^3$)	24-Hour SO_2 Average .8% S Q=3.58 g/sec ($\mu g/m^3$)	24-Hour SO_2 Average .3% S Q=1.34 g/sec ($\mu g/m^3$)
22 Unit Cluster						
.1	1.8	0.6	2672.0	1002.0	501.0	188.0
.2	1.0	0.6	1850.0	694.0	506.0	190.0
.5	0.2	0.3	695.0	261.0	276.0	164.0
2.0	0.3	0.1	162.0	60.7	141.0	157.0
20.0	—	—	5.4	2.0	5.8	2.2
5 Unit Cluster						
.1	0.5	0.1	589.0	221.0	99.2	37.2
.2	0.3	0.1	440.0	165.0	114.0	43.0
.5	0.1	—	165.0	62.0	64.5	24.2
2.0	—	—	37.5	14.0	32.3	12.1
20.0	—	—	1.2	0.4	1.3	0.5

— Concentrations less than 0.1

Table E - 7

GENERAL ELECTRIC
MS5001P; DF-2 FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	.3% S Q=11.8g/sec	1.0% S Q=39.3g/sec	.3% S Q=11.8g/sec	1.0% S Q=39.3g/sec	.3% S Q=11.8g/sec	1.0% S Q=39.3g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	131.0	435.0	41.1	137.0	1.2	4.1
.2	62.3	208.	16.1	53.6	0.3	1.3
.5	28.9	96.3	6.3	21.1	0.1	0.5
2.0	7.0	23.6	2.1	7.0	0.1	0.5
20.0	4.4	24.7	1.3	4.4	0.1	0.5

	1-Hour CO Average 250 ppm Q = 24. g/sec	8-Hour CO Average 250 ppm Q = 24. g/sec	Annual NO_x Average 147 ppm Q = 23.5 g/sec
	(mg/m^3)	(mg/m^3	($\mu g/m^3$)
.1	0.3	0.2	2.5
.2	0.2	0.1	0.8
.5	0.1	—	1.2
2.0	—	—	0.3
20.0	—	—	0.3

— Concentrations less than 0.1

Table E - 8

GENERAL ELECTRIC

MS5001P; DF-2 FUEL

Maximum Ground Level Concentration; Dallas, Texas, 1964

4, 8, and 16 Unit Cluster (see Figure 6-1)

Downwind Distance	1-Hour CO Average	8-Hour CO Average	3-Hour SO_2 Average		24-Hour SO_2 Average	
	250 ppm Q=24 g/sec	250 ppm Q=24 g/sec	1.0% S Q=39.3 g/sec	.3% S Q=11.8 g/sec	1.0% S Q=39.3 g/sec	.3% S Q=11.8 g/sec
(km)	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
	16 Units					
.1	2.1	1.3	2454.0	737.0	926.0	298.0
.2	1.1	0.6	1373.0	412.0	446.0	134.0
.5	0.7	0.4	899.0	270.0	266.0	80.0
2.0	0.3	0.1	339.0	102.0	113.0	34.0
20.0	0.0	0.0	38.5	11.5	21.0	6.3
	8 Units					
.1	2.1	1.2	2444.0	733.0	755.0	227.0
.2	1.1	0.6	1175.0	353.0	352.0	106.0
.5	0.4	0.2	470.0	141.0	140.0	42.0
2.0	0.1	0.1	164.0	49.0	54.1	16.3
20.0	—	—	19.1	5.7	10.5	3.1
	4 Units					
.1	1.2	0.7	1457.0	438.0	454.0	136.0
.2	0.6	0.3	648.2	195.0	195.0	59.0
.5	0.2	0.1	244.0	73.0	43.0	13.0
2.0	0.0	0.0	81.0	50.0	27.0	8.1
20.0	0.0	0.0	9.6	2.9	5.3	1.6

— Concentrations less than 0.1

Table E - 9

GENERAL ELECTRIC

MS5001P; DF-2 FUEL

(Spinning Reserve Mode)

Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	.3% S Q=3.93g/sec	1.0% S Q=13.1g/sec	.3% S Q=3.93g/sec	1.0% S Q=13.1g/sec	.3% S Q=3.93g/sec	1.0% S Q=13.1g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	89.8	299.	32.6	108.6	2.2	7.4
.2	44.1	147.	14.8	49.2	0.8	2.8
.5	17.2	57.4	5.4	18.1	0.9	3.3
2.0.	6.0	20.2	3.3	11.2	0.3	1.1
20.0	4.0	13.3	0.9	3.1	0.6	.2

	1-Hour CO Average 500 ppm Q = 43.1 g/sec	8-Hour CO Average 500 ppm Q = 43.1 g/sec
	(mg/m^3)	(mg/m^3)
.1	1.3	0.8
.2	0.6	0.4
.5	0.2	0.1
2.0	0.1	0.1
20.0	0.1	0.0

— Concentrations less than 0.1

Table 10

GENERAL ELECTRIC
MS5001P; DF-2 FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964
16 Unit Cluster (see Figure 6-1)

Spinning Reserve (SR)
Regular Model (RM)

Downwind Distance (km)	1-Hour CO Average SR; 500 ppm Q=43.1g/sec RM; 7.8 ppm Q=0.75 g/sec (mg/m^3)	8-Hour CO Average SR; 500 ppm Q=43.1g/sec RM; 7.8 ppm Q=0.75 g/sec (mg/m^3)	3-Hour SO_2 Average SR; 1.0% S Q=13.1 g/sec RM; 1.0% S Q=39.3g/sec (µg/m^3)	SR; .3% S Q=3.93 g/sec RM; .3% S Q=11.8 g/sec (µg/m^3)	24-Hour SO_2 Average SR; 1.0% Q=13.1 g/sec RM; 1.0% S Q=39.3g/sec (µg/m^3)	SR; .3% S Q=3.93 g/sec RM; .3% S Q=11.8 g/sec (µg/m^3)
	4 Units SR 12 Units RM					
.1	4.5	2.9	1946.0	584.3	839.0	252.0
.2	2.3	1.4	1189.0	357.0	423.0	127.0
.5	0.9	0.6	831.0	250.0	258.0	77.0
2.0	0.3	0.2	316.0	95.0	112.0	34.0
20.0	—	—	32.0	9.6	18.0	5.3
	8 Units SR 8 Units RM					
.1	7.8	5.0	1770.0	531.0	793.0	238.0
.2	4.2	2.6	1116.0	335.0	407.0	122.0
.5	1.2	1.1	796.0	239.0	250.0	75.0
2.0	0.6	0.4	301.0	93.0	111.0	33.0
20.0	—	—	26.1	7.8	15.0	4.4
	16 Units SR 0 Units RM					
.1	7.8	5.6	1770.0	531.0	811.0	243.0
.2	4.4	2.9	1116.0	335.0	432.0	129.0
.5	2.7	1.8	795.0	239.0	244.0	73.0
2.0	1.1	0.8	259.0	78.0	111.0	33.4
20.0	0.1	—	14.0	4.2	7.8	2.3

— Concentrations less than 0.1

Table E-11

GENERAL ELECTRIC
MS5002B; DF-2 FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	1.0% S Q=39.3g/sec	.3% S Q=39.3g/sec	1.0% S Q=39.3g/sec	.3% S Q= 11.8g/sec	1.0% S Q=39.3g/sec	.3% S Q=11.8g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	1393.0	418.0	473.0	142.0	23.9	7.2
.2	453.0	136.0	144.0	43.2	5.5	1.6
.5	148.0	44.4	37.8	11.3	1.5	0.4
2.0	38.4	11.5	14.8	4.4	1.6	0.5
20.0	38.7	11.6	6.6	1.9	—	—

	1-Hour CO Average 250 ppm Q = 24. g/sec	8-Hour CO Average 250 ppm Q = 24. g/sec	Annual NO_x Average 147 ppm Q = 23.5 g/sec
	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)
.1	1.1	0.7	14.3
.2	0.4	0.2	3.3
.5	0.1	0.1	0.9
2.0	0.4	—	0.9
20.0	—	—	—

— Concentrations less than 0.1

Table E - 12

GENERAL ELECTRIC

MS5002B; DF-2 FUEL

Maximum Ground Level Concentration; Dallas, Texas, 1964

8,4 and 2 Unit Cluster (see Figure 5-1)

Downwind Distance	1-Hour CO Average	8-Hour CO Average	3-Hour SO_2 Average		24-Hour SO_2 Average	
	250 ppm Q=24 g/sec	250 ppm Q= 24 g/sec	1.0% S Q= 39.3 g/sec	.3% S Q= 11.8 g/sec	1.0% S Q=39.3 g/sec	.3% S Q= 11.8 g/sec
(km)	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
	8 Units					
.1	5.9	3.6	7255.0	2177.0	2400.0	720.0
.2	2.4	1.4	2860.0	858.0	900.0	270.0
.5	0.7	0.4	868.0	260.0	268.0	80.4
2.0	0.2	0.1	223.0	87.9	105.0	31.6
20.0	0.0	0.0	22.5	6.7	12.5	3.7
	4 Units					
.1	3.6	2.2	4500.0	1350.0	1527.0	458.0
.2	1.3	0.8	1619.0	486.0	512.0	154.0
.5	0.4	0.2	454.0	136.0	141.0	42.2
2.0	—	0.1	147.0	43.9	52.7	15.8
20.0	—	—	11.2	3.3	6.2	1.8
	2 Units					
.1	2.1	1.3	2570.0	771.0	869.0	261.0
.2	0.7	0.4	869.0	261.0	275.0	82.5
.5	0.2	0.1	233.0	69.9	72.1	21.6
2.0	—	—	73.4	22.0	26.4	7.9
20.0	—	—	5.6	1.6	3.1	0.9

— Concentrations less than 0.1

Table E - 13

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Simple Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	.3% S Q=26.5g/sec	1.0% S Q=88.2g/sec	.3% S Q=26.5g/sec	1.0% S Q=88.2g/sec	.3% S Q=26.5g/sec	1.0% S Q=88.2g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	213.0	709.0	60.6	202.0	0.3	1.1
.2	57.5	192.0	15.5	51.6	0.0	0.2
.5	28.1	94.0	4.7	15.7	0.0	0.1
2.0	10.4	35.0	1.4	4.7	0.0	0.1
20.0	10.4	35.0	2.1	7.0	0.2	0.8

	1-Hour CO Average 250 ppm Q=43.6g/sec	8-Hour CO Average 250 ppm Q=43.6g/sec	Annual NO_x Average 163 ppm Q=46.7g/sec
	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)
.1	0.5	0.3	0.5
.2	0.1	0.1	0.1
.5	0.0	—	0.0
2.0	0.0	—	—
20.0	0.0	—	0.4

— Concentrations less than 0.

Table E - 14

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Simple Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964
4, 8, and 16 Unit Cluster (see Figure 6-1)

Downwind Distance	1-Hour CO Average	8-Hour CO Average	3-Hour SO_2 Average		24-Hour SO_2 Average	
	250 ppm Q=43.6 g/sec	250 ppm Q=43.6 g/sec	1.0% S Q=88.2 g/sec	.3% S Q=26.5 g/sec	1.0% S Q=88.2 g/sec	.3% S Q=26.5 g/sec
(km)	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
	16 Units					
.1	2.1	1.0	2586.0	776.0	708.0	213.0
.2	1.0	0.4	1217.0	365.0	323.0	96.9
.5	0.5	0.2	525.0	157.0	139.0	41.7
2.0	0.1	0.1	139.0	41.6	38.1	11.4
20.0	0.1	—	76.5	22.9	37.9	11.4
	8 Units					
.1	1.6	0.8	2118.0	635.0	585.0	176.0
.2	0.7	0.3	860.0	258.0	229.0	68.7
.5	0.3	0.1	314.0	94.3	83.0	24.9
2.0	0.1	0.0	32.0	9.6	16.6	4.9
20.0	—	0.0	38.5	11.6	19.1	5.7
	4 Units					
.1	1.2	0.6	1627.0	488.0	454.0	136.0
.2	0.5	0.2	563.0	169.0	151.0	45.2
.5	0.2	0.1	175.0	52.6	46.2	13.9
2.0	—	—	28.8	8.6	7.6	2.3
20.0	—	—	19.3	5.8	9.5	2.8

Table E - 15

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Simple Cycle
(Spinning Reserve Mode)
Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	1.0% S Q=29.4g/sec	.3% S Q=8.83g/sec	1.0% S Q=29.4g/sec	.3% S Q=8.83g/sec	1.0% S Q=29.4g/sec	.3% S Q=8.83g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	591.0	177.0	194.0	58.1	7.5	2.2
.2	209.0	62.7	63.1	18.9	1.9	0.6
.5	82.3	24.7	19.3	5.7	0.6	0.2
2.0	20.6	6.1	6.0	1.8	0.5	0.2
20.0	20.0	6.1	3.6	1.1	0.4	0.1

	1-Hour CO Average 500 ppm Q = 84.7 g/sec	8-Hour CO Average 500 ppm Q = 84.7 g/sec
	(mg/m^3)	(mg/m^3)
.1	2.3	1.4
.2	0.8	0.5
.5	0.3	0.2
2.0	0.1	—
20.0	0.1	—

— Concentrations less than 0.1

Table E 16

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Simple Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964
16 Unit Cluster (see Figure 6-1)

Spinning Reserve (SR)
Regular Model (RM)

Downwind Distance (km)	1-Hour CO Average SR; 500 ppm Q=84.8 g/sec RM; 2.5 ppm Q=0.39 g/sec (mg/m^3)	8-Hour CO Average SR; 500 ppm Q=84.8 g/sec RM; 2.5 ppm Q=0.39 g/sec (mg/m^3)	3-Hour SO_2 Average SR; 1.0% S Q=29.4 g/sec RM; 1.0% S Q=88.2 g/sec (µg/m^3)	3-Hour SO_2 Average SR; .3% S Q=8.8 g/sec RM; .3%S Q=26.5 g/sec (µg/m^3)	24-Hour SO_2 Average SR; 1.0% S Q=29.4 g/sec RM; 1.0% S Q=88.2 g/sec (µg/m^3)	24-Hour SO_2 Average SR; .3% S Q=8.8 g/sec RM; .3% S Q=26.5 g/sec (µg/m^3)
	4 Units SR 12 Units RM					
.1	5.9	3.6	2447.0	734.0	731.0	220.0
.2	2.6	1.5	1281.0	384.0	365.0	109.0
.5	0.9	0.5	564.0	169.0	158.0	47.3
2.0	0.2	0.2	178.0	53.5	53.3	16.0
20.0	—	0.0	64.6	19.4	32.4	9.7
	8 Units SR 8 Units RM					
.1	8.3	4.9	2514.0	754.0	770.2	231.0
.2	4.2	2.3	1342.0	403.0	395.0	118.0
.5	1.7	0.9	593.0	178.0	172.0	51.5
2.0	0.6	0.3	214.0	64.3	67.6	20.3
20.0	0.1	—	52.7	15.8	26.9	8.0
	16 Units SR 0 Units RM					
.1	10.8	6.3	2615.0	785.0	819.0	246.0
.2	6.1	3.4	1421.0	426.0	432.0	130.0
.5	2.8	1.5	636.0	190.4	192.0	58.0
2.0	1.2	0.7	274.0	82.3	93.6	28.1

— Concentrations less than 0.1

Table E - 17

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Unfired Combined Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964

Main Stack (MS)
By-Pass Stack (BP)

Downwind Distance	3-Hour SO_2 Average 1% S Q_{MS}=88.2 g/sec Q_{BP}=.882 g/sec	24-Hour SO_2 Average 1% S Q_{MS}=88.2 g/sec Q_{BP}=.882 g/sec	Annual SO_2 Average 1% S Q_{MS}=88.2 g/sec Q_{BP}=.882 g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	85.5	58.2	7.1
.2	82.5	29.9	3.5
.5	121.0	25.3	2.3
2.0	41.2	10.2	1.1
20.0	34.9	7.6	1.0

	1-Hour CO Average 250 ppm Q_{MS}=43.6 g/sec Q_{BP}=.436 g/sec	8-Hour CO Average 250 ppm Q_{MS}=43.6 g/sec Q_{BP}=.436 g/sec	Annual NO_x Average 163 ppm Q_{MS}=46.7 g/sec Q_{BP}=.467 g/sec
	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)
.1	0.1	—	3.7
.2	0.1	—	1.8
.5	0.1	—	1.2
2.0	—	—	0.6
20.0	—	—	0.5

Table E - .8

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Unfired Combined Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964
16 Unit Cluster (see Figure 6-1)

Main Stack (MS)
By-Pass Stack (BP)

Downwind Distance	1-Hour CO Average 250 ppm Q_{MS}=43.6g/sec Q_{BP}=.436g/sec	8-Hour CO Average 250 ppm Q_{MS}=43.6g/sec Q_{BP}=.436g/sec	3-Hour SO_2 Average 1% S Q_{MS}=88.2g/sec Q_{BP}=.882g/sec	24-Hour SO_2 Average 1% S Q_{MS}=88.2g/sec Q_{BP}=.882g/sec
(km)	(mg/m^3)	(mg/m^3)	(μg/m^3)	(μg/m^3)
.1	0.7	0.3	842.0	427.0
.2	0.7	0.2	1049.0	322.0
.5	0.6	0.1	843.0	200.0
2.0	0.2	0.1	301.0	89.4
20.0	—	—	32.4	50.7

— Concentrations less than 0.1

Table E-19

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Regenerative Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	.3% S Q=19.33 g/sec	1.0% S Q=64.4 g/sec	.3% S Q=19.33 g/sec	1.0% S Q=64.4 g/sec	.3% S Q=19.33 g/sec	1.0% S Q=64.4 g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	474.0	1581.0	165.0	550.0	7.9	26.5
.2	168.0	561.0	85.8	286.0	2.0	6.8
.5	80.6	269.00	16.2	53.9	0.6	2.0
2.0	15.4	51.4	5.6	18.7	0.5	1.8
20.0	15.1	50.5	2.5	8.6	0.2	0.9

	1-Hour CO Average 250 ppm Q=42.4 g/sec	8-Hour CO Average 250 ppm Q= 42.4 g/sec	Annual NO_x Average 373 ppm Q=117.9 g/sec
	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)
.1	1.3	0.9	49.9
.2	0.6	0.3	12.8
.5	0.2	0.1	3.8
2.0	0.1	—	3.4
20.0	0.1	—	1.8

— Concentrations less then 0.1

Table E-20

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Regenerative Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964
4,8, and 16 Unit Cluster (see Figure

Downwind Distance	1-Hour CO Average	8-Hour CO Average	3-Hour SO_2 Average		24-Hour SO_2 Average	
	250 ppm	250 ppm	1.0% S	.3% S	1.0% S	.3% S
	Q=42.4 g/sec	Q=42.4 g/sec	Q=64.4 g/sec	Q=19.3 g/sec	Q=64.4 g/sec	Q=19.3 g/sec
(km)	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
	16 Units					
.1	6.7	4.2	8200.0	2460.0	2792.0	838.0
.2	4.2	2.5	4779.0	1434.0	1525.0	458.0
.5	1.9	1.0	2020.0	606.0	637.0	191.0
2.0	0.7	0.5	781.0	234.0	284.0	85.1
20.0	0.1	—	66.1	19.8	36.8	11.0
	8 Units					
.1	4.6	2.9	5874.0	1762.0	1617.0	485.0
.2	2.7	1.6	3112.0	934.0	1000.0	300.0
.5	1.1	0.6	1192.0	358.0	366.0	110.0
2.0	0.4	0.2	396.0	118.0	142.0	43.4
20.0	—	—	33.3	9.9	18.5	5.5
	4 Units					
.1	2.8	1.9	3782.0	1135.0	1357.0	407.0
.2	1.6	0.9	1857.0	557.0	602.0	181.0
.5	0.6	0.3	640.0	192.0	199.0	59.7
2.0	0.2	0.1	198.0	59.4	70.7	21.2
20.0	—	—	16.7	5.0	9.2	2.7

— Concentrations less than 0.1

Table E-21

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Regenerative Cycle
(Spinning Reserve Mode)
Maximum Ground Level Concentration

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	.3% S Q=6.44g/sec	1.0% S Q=21.5g/sec	.3% S Q=6.44g/sec	1.0% S Q=21.5g/sec	.3% S Q=6.44g/sec	1.0% S Q=21.5g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	304.0	1014.0	163.0	543.0	17.5	41.5
.2	121.0	402.0	54.2	181.0	4.0	13.3
.5	38.4	128.3	15.6	52.0	1.2	4.0
2.0	15.8	52.9	9.4	31.4	1.0	3.3
20.0	7.1	23.9	1.4	4.9	0.1	0.4

	1-Hour CO Average	8-Hour CO Average
	500 ppm Q=84.8g/sec	500 ppm Q=84.8g/sec
	(mg/m^3)	(mg/m^3)
.1	5.2	3.4
.2	2.1	1.4
.5	0.7	0.4
2.0	0.3	0.1
20.0	0.3	0.1

—— Concentrations less than 0.1

Table 2

GENERAL ELECTRIC
MS7001B; DF-2 FUEL
Regenerative Cycle
Maximum Ground Level Concentration; Dallas, Texas, 1964
16 Unit Cluster (see Figure 6-1)

Spinning Reserve (SR)
Regular Mode (RM)

Downwind Distance (km)	1-Hour CO Average SR; 500 ppm Q=84.8 g/sec RM; 5 ppm Q=.97 g/sec (mg/m^3)	8-Hour CO Average SR; 500 ppm Q=84.8 g/sec RM; 5 ppm Q=.97 g/sec (mg/m^3)	3-Hour SO_2 Average SR; 1.0% S Q=21.5 g/sec RM; 1.0% S Q=64.4 g/sec ($\mu g/m^3$)	3-Hour SO_2 Average SR; .3% S Q=6.44 g/sec RM; .3% S Q=19.3 g/sec ($\mu g/m^3$)	24-Hour SO_2 Average SR; 1.0% S Q=21.5 g/sec RM; 1.0% Q=64.4 g/sec ($\mu g/m^3$)	24-Hour SO_2 Average SR; .3% S Q=6.44 g/ RM; .3% S Q=19.3 g/ ($\mu g/m^3$)
	4 Units SR 12 Units RM					
.1	10.5	7.6	6844.0	2053.0	2501.0	750.0
.2	6.4	4.3	4240.0	1272.0	1436.0	431.0
.5	2.5	1.6	1856.0	557.0	611.0	183.0
2.0	0.7	0.5	716.4	215.0	275.0	82.5
20.0	—	—	5.2	16.6	30.9	9.2
	8 Units SR 8 Units RM					
.1	17.4	12.4	6217.0	1865.0	2395.0	719.0
.2	11.0	7.4	3906.0	1172.0	1385.0	416.0
.5	4.5	2.9	1716.0	515.0	588.0	176.0
2.0	1.3	1.0	649.0	195.0	264.0	79.2
20.0	0.1	0.1	44.5	13.4	25.0	7.5
	16 Units SR 0 Units RM					
.1	26.1	18.2	5592.0	1678.0	2299.0	689.0
.2	17.4	11.6	3478.0	1043.0	1687.0	506.0
.5	7.9	5.1	1484.0	445.0	547.0	164.0
2.0	2.6	1.9	505.0	151.0	237.0	71.1
20.0	0.1	0.1	23.2	6.9	13.4	4.0

— Concentrations less than 0.1

Table E-23

GENERAL ELECTRIC
MS7001C; DF-2 FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	.3% S Q=27.8g/sec	1.0% S Q=92.3g/sec	.3%S Q=27.8g/sec	1.0% S Q=92.3g/sec	.3% S Q=27.3g/sec	1.0% S Q=92.3g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	175.0	584.0	29.6	98.8	0.1	0.3
.2	44.3	148.0	7.4	24.7	—	0.1
.5	48.5	162.0	1.9	6.3	—	0.1
2.0	10.9	36.3	1.4	4.7	—	0.1
20.0	10.2	34.0	2.1	6.9	0.2	0.8

	1-Hour CO Average 250 ppm Q=60.7g/sec	8-Hour CO Average 250 ppm Q=60.7g/sec	Annual NO_x Average 164 ppm Q=66.9g/sec
	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)
.1	0.6	0.2	0.2
.2	0.2	—	—
.5	0.1	—	—
2.0	—	—	0.1
20.0	—	—	0.5

— Concentrations less than 0.1

Table E-4

GENERAL ELECTRIC
MS7001C; DF-2 FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964
4,8 and 16 Unit Cluster (see Figure 6-1)

Downwind Distance	1-Hour CO Average	8-Hour CO Average	3-Hour SO_2 Average		24-Hour SO_2 Average	
	250 ppm Q=60.7 g/sec	250 ppm Q=60.7 g/sec	1.0% S Q=92.3 g/sec	.3% S Q=27.8 g/sec	1.0% S Q=92.3 g/sec	.3% S Q=27.8 g/sec
(km)	(mg/m^3)	(mg/m^3)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
	16 Units					
.1	2.1	0.7	2021.0	606.0	342.0	103.0
.2	1.0	0.3	910.0	273.0	156.0	46.7
.5	0.5	0.1	387.0	116.0	71.7	21.5
2.0	0.1	—	79.8	23.9	18.7	5.6
20.0	0.1	—	76.3	22.9	36.7	11.0
	8 Units					
.1	1.7	0.5	1676.0	503.0	282.0	84.6
.2	0.8	0.2	647.0	194.0	109.0	32.7
.5	0.3	0.1	232.0	69.4	41.6	12.5
2.0	—	—	34.3	10.3	4.8	1.4
20.0	—	—	38.4	11.5	18.5	5.5
	4 Units					
.1	1.3	0.4	1306.0	392.0	220.0	65.9
.2	0.5	0.1	427.0	128.0	71.7	21.5
.5	0.1	—	129.0	38.7	22.8	6.8
2.0	—	—	15.9	4.7	3.5	1.0
20.0	—	—	19.2	5.7	9.2	2.7

— Concentrations less than 0.1

Table E-25

GENERAL ELECTRIC
MS7001C; DF-2 FUEL
(Spinning Reserve Mode)
Maximum Ground Level Concentration; Dallas, Texas, 1964

Downwind Distance	3-Hour SO_2 Average		24-Hour SO_2 Average		Annual SO_2 Average	
	.3% S Q=9.3g/sec	1.0% S Q=30.9g/sec	.3% S Q=9.3g/sec	1.0% S Q=30.9g/sec	.3% S Q=9.3g/sec	1.0% S Q=30.9g/sec
(km)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)	($\mu g/m^3$)
.1	151.0	500.0	47.3	158.0	0.8	2.6
.2	51.7	172.0	14.7	49.0	0.2	0.7
.5	21.4	71.3	4.4	14.7	0.1	0.3
2.0	4.7	15.9	1.2	4.1	0.1	0.3
20.0	5.2	17.6	1.0	3.3	0.1	0.4

	1-Hour CO Average 500 ppm Q = 121.4 g/sec	8-Hour CO Average 500 ppm Q = 121.4 g/sec
	(mg/m^3)	(mg/m^3)
.1	2.7	1.6
.2	0.9	0.5
.5	0.4	0.2
2.0	0.1	——
20.0	0.1	——

—— Concentrations less than 0.1

Tabl(-26

GENERAL ELECTRIC
MS7001C; DF-2 FUEL
Maximum Ground Level Concentration; Dallas, Texas, 1964
4,8 and 16 Unit Cluster (see Figure 6-1)

Spinning Reserve (SR)
Regular Mode (RM)

Downwind Distance	1-Hour CO Average SR; 500 ppm Q=121.4 g/sec RM; 2.1 ppm Q=0.51 g/sec	8-Hour CO Average SR; 500 ppm Q=121.4 g/sec RM; 2.1 ppm Q=0.51 g/sec	3-Hour SO_2 Average SR; 1.0% S Q=30.8 g/sec RM; 1.0% S Q=92.3 g/sec	 SR; .3% S Q=9.3 g/sec RM; .3% S Q=27.8 g/sec	24-Hour SO_2 Average SR; 1.0% Q=30.8 g/sec RM; 1.0% S Q=92.3 g/sec	 SR; .3% S Q=9.3 g/s RM; .3% S Q=27.8 g/
(km)	(mg/m^3)	(mg/m^3)	(μg/m^3)	(μg/m^3)	(μg/m^3)	(μg/m^3)
	4 Units SR 12 Units RM					
.1	6.9	4.0	1947.0	584.0	501.0	150.0
.2	2.9	1.5	980.0	294.0	231.0	69.2
.5	1.0	0.5	425.0	128.0	97.5	29.3
2.0	0.3	0.2	113.0	33.9	30.5	9.1
20.0	—	—	64.6	19.4	31.5	9.4
	8 Units SR 8 Units RM					
.1	9.5	5.4	2018.0	605.0	567.0	170.0
.2	4.6	2.4	1039.0	312.0	275.0	82.4
.5	1.8	0.9	454.0	136.0	117.0	35.1
2.0	0.6	0.3	146.0	43.7	42.5	12.7
20.0	0.1	0.1	52.9	15.9	26.3	7.9
	16 Units SR 0 Units RM					
.1	12.3	6.8	2116.0	635.0	637.0	191.0
.2	6.8	3.5	1116.0	335.0	327.0	98.1
.5	3.1	1.5	496.0	149.0	145.0	43.4
2.0	1.2	0.7	204.0	61.3	65.5	19.7
20.0	0.2	0.1	29.9	8.9	16.1	4.8

— Concentrations less than 0.1

APPENDIX F. ENFORCEMENT ASPECTS

Factors which affect gas turbine emissions are changes in power output, fuel consumption, fuel analysis, ambient conditions (temperature, pressure and humidity), and water or steam injection rate. The effects of these variables have been discussed in Chapter 3.

Test method, "Determination of Nitrogen Oxide and Oxygen from Stationary Gas Turbines", specifies test methods and procedures for both field and laboratory measurements.

The compliance test for stationary gas turbines requires the use of instruments to measure O_2 and NO_x. Instruments were specified because they can accurately measure low concentrations of pollutants and are best suited for production line testing. For repetitive emission tests, such as during engine performance testing, the stack can be designed so that one sample point will be representative.

Compliance testing of gas turbines using the methods specified in the regulation should be straightforward. The instruments may be based on any of several analytical techniques which meet the performance criteria specified in the test method. Possible problems which may be encountered are instrument malfunctions and stack configurations which are not amenable to reproducible sampling.

Testing of turbines may present a special problem which requires an interpretation of the method by the enforcement agency. EPA's reference test method specifies that emissions from the turbine shall be sampled downstream of the power turbine. Furthermore, for combined cycle plants which are supplementally fired, sampling shall be performed between the power turbine and the boiler (this excludes emissions from the supplemental firing). In turbines which are ducted to a common stack, emissions shall be measured in the individual breachings before the common stack. Less satisfactory alternatives such as testing at the top of the stack, upstream of the power turbine, or at a single point, should be avoided.

Modifications which may be necessary to permit emission testing may include installation of an extension to the stack or temporary removal of noise silencers. Any modification should be reviewed by the enforcement agency to insure that it does not result in an unrepresentative sample.

Although emissions from a given gas turbine are reportedly almost constant, they increase markedly with a decrease in overall efficiency. A decrease in efficiency can be caused by degradation of the combustor, erosion of the compressor or turbine blades, or deposits on the compressor or turbine blades. For large turbines used to generate peaking power and burning distillate fuel, the rate of efficiency decrease has been estimated at about one percent per year.[1] Changes in efficiency can be monitored by standard process instrumentation, such as pressure ratios, exhaust temperatures, and fuel flow. Combustor life can be estimated so that maintenance activities and emission measurement tests can be scheduled accordingly. Deposits of material on the compressor and turbine surfaces are periodically removed by methods such as steam cleaning or by injecting crushed walnut shells or rice hulls into the air intake.

Compliance tests to verify emissions must be performed after each major repair and when combustor sections are modified or replaced. If the process instrumentation indicates that a significant degradation in turbine efficiency has occurred and engine maintenance or overhaul are not scheduled for the near future, a compliance test should be performed to determine compliance, and if necessary, provide a basis for specifying mandatory maintenance to reduce emissions.

Formation of NO_x from organic or fuel-bound nitrogen is not retarded by wet control methods, but is retarded by some dry control techniques as discussed in Chapter 4. The performance test must be performed while the turbine is using a fuel which contains the maximum organic nitrogen which will be used at the installation.

The recommended emission levels permit the actual NO_x emission rate to be adjusted based on the efficiency of the gas turbine. Those with greater efficiencies (greater work per unit of fuel consumed) typically operate at higher temperatures and pressures. This is conducive to NO_x formation and higher concentrations of NO_x are emitted.

Correction factors to adjust NO_x emissions to a reference humidity and pressure have been recommended. If a manufacturer has developed correction factors for humidity and pressure specifically for his model of turbine, he may use those correction factors after submittal of substantiating data to EPA and approval by EPA. Correction factors for temperature are optional and, if used, must be developed by the manufacturer for the specific model. The International Standards Organization (ISO) standard day conditions of one atmosphere, 59°F and 60 percent relative humidity are chosen as the reference conditions. Since the correction factors recommended were developed by manufacturers for turbines which use conventional combustors operating at, or near, stoichiometric conditions, they cannot be applied to turbines which use emerging technology combustors such as the low emission lean-burn types (described in Chapter 4) which operate at off-stoichiometric conditions. Correction factors for turbines which use non-conventional combustors must be developed by the manufacturer and approved by EPA for each specific turbine model.

The supplementary fired portion of a combined cycle gas turbine, when fired with a fossil fuel at a heat input of greater than 250 million Btu per hour, must meet the standards of performance for fossil fuel-fired steam generators. The gas turbine portion of all combined cycle units shall meet the standards of performance for gas turbines.

Since the maximum emissions of NO_x from gas turbines occur at full load, compliance tests must be performed at full load. Since NO_x emissions will vary with changes in ambient humidity and pressure (as discussed in Chapter 3), gas turbine owners using water or steam injection to meet the recommended emission level referenced to ISO conditions will have to vary the water flowrates commensurately. An alternative to this variation of water flowrate is to use a constant water flowrate which will provide a reduction in uncontrolled NO_x emissions to meet the recommended level under the worst anticipated conditions of ambient humidity and pressure.

In arctic and subarctic areas, the "ice fog" which sometimes forms may result in reduced visibility and traffic problems (see Chapter 4 for discussion). The regulation permits suspension of the use of wet control methods on those occasions when the resulting ice fog is deemed a visibility hazard by an authorized representative of the state or local government agency responsible for traffic safety.

REFERENCES FOR APPENDIX F

1. Letter from Dibelius, N. R., Vice Chairman of the Combustion and Fuels Committee, Gas Turbine Division of the ASME, to Steigerwald, B. J., OAQPS, EPA. p. 5. May 1, 1973.

APPENDIX - G

METHOD 20--DETERMINATION OF NITROGEN OXIDES, SULFUR DIOXIDE, AND OXYGEN EMISSIONS FROM STATIONARY GAS TURBINES

1. Principle and Applicability

1.1 Principle. A gas sample is continuously extracted from the exhaust stream of a stationary gas turbine; a portion of the sample stream is conveyed to instrumental analyzers for determination of nitrogen oxides (NO_x) and oxygen (O_2) content. During each NO_x and O_2 determination, a separate measurement of sulfur dioxide (SO_2) emissions is made, using Method 6, or its equivalent. The O_2 determination is used to adjust the NO_x and SO_2 to a reference condition.

1.2 Applicability. This method is applicable for the determination of nitrogen oxide, sulfur dioxide, and oxygen emissions from stationary gas turbines. For the NO_x and O_2 determinations, this method includes: (1) measurement system design criteria, (2) analyzer performance specifications and performance test procedures; and (3) procedures for emission testing.

2. Apparatus and Reagents

2.1 Measurement System. The equipment required to extract, transport, and analyze the gas sample constitutes the "measurement system." A schematic of the measurement system is shown in Figure 20-1. (Measurement system performance specifications are described in detail in Section 3.) The essential components of the measurement system are described below.

2.1.1 Probe. Stainless steel type 316 or equivalent, to transport gas from stack.

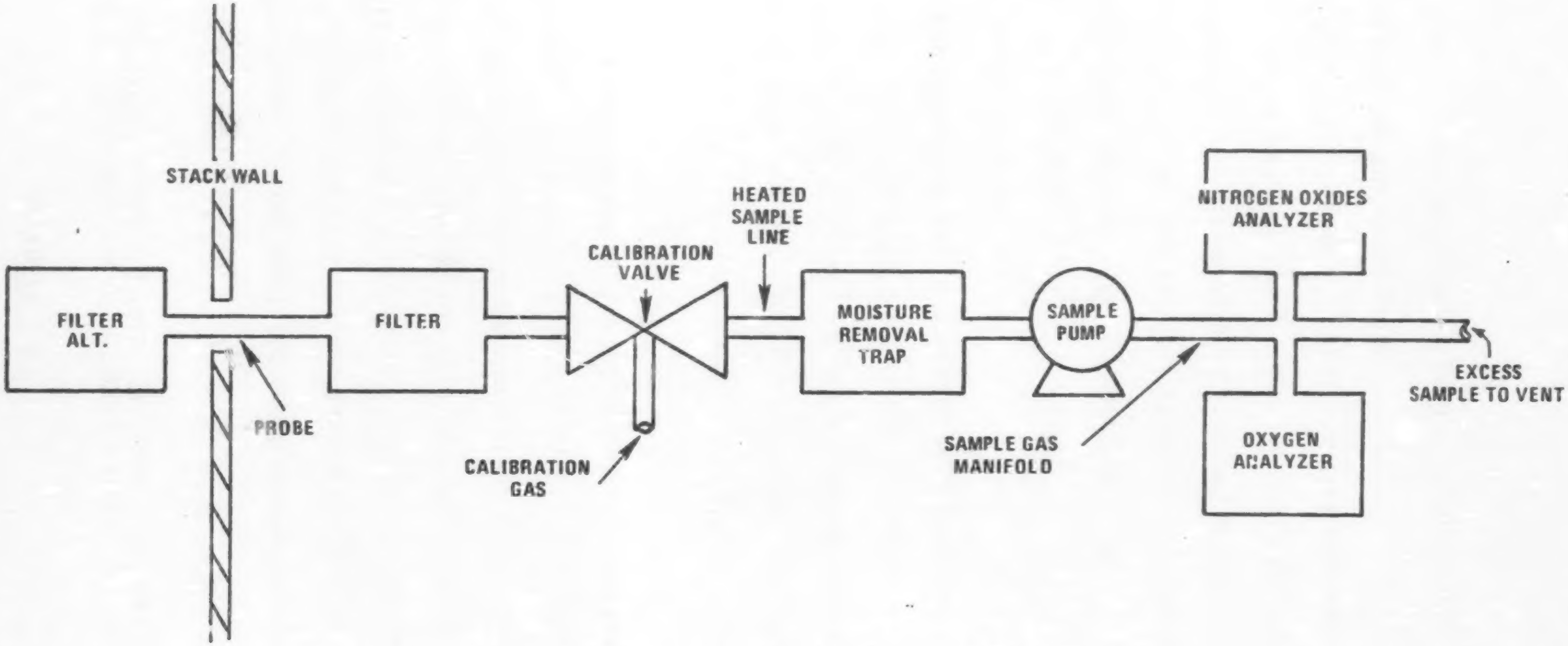
STACK WALL
FILTER ALT.
PROBE
FILTER
CALIBRATION VALVE
CALIBRATION GAS
HEATED SAMPLE LINE
MOISTURE REMOVAL TRAP
SAMPLE PUMP
SAMPLE GAS MANIFOLD
NITROGEN OXIDES ANALYZER
OXYGEN ANALYZER
EXCESS SAMPLE TO VENT

Figure 20.1 Measurement system design for stationary gas turbine tests.

2.1.2 Particulate Filter. A filter is used to remove particulates ahead of the calibration valve assembly. In most cases, either in-stack or out-stack filter location is acceptable; however, out-stack filtration is required when the sample gas temperature is above 500°C (930°F). The filtration temperature shall be at least 120°C (250°F) to prevent moisture condensation. Glass fiber filters, of the type specified in EPA Method 5, or equivalent, are recommended.

2.1.3 Calibration Valve Assembly. A three-way valve assembly is used to direct the zero and span calibration gases to the analyzers. This assembly shall be located directly behind the probe and filter and shall be capable of blocking the sample gas flow and introducing the span and zero gases when the system is in the calibration mode.

2.1.4 Calibration Gases. Calibration gases are used to perform zero, span and calibration checks of the analyzers during each test run. The concentrations and specifications of these gases are described in detail in Sections 2.2 and 6.2.

2.1.5 Heated Sample Line. A FEP fluorocarbon or stainless steel (type 316 or equivalent) sample line is used to transport the gases to the sample conditioner and analyzers. The sample gas shall be maintained at least 5°C (10°F) above the stack gas dew point to prevent moisture condensation.

2.1.6 Moisture Trap. A moisture trap, designed to reduce the dew point of the sample gas to 3°C (37°F) or less, is used. For instruments not affected by water vapor, this device is not required; however, the moisture content shall be determined using methods subject to the approval of the Administrator and the NO_x and O_2 concentrations shall be corrected to a dry gas basis.

2.1.7 Pump. A nonreactive leak-free sample pump is used to pull the sample gas through the system at a flow rate sufficient to minimize transport delay. The pump shall be made from or coated with nonreactive material (FEP fluorocarbon or type 316 stainless steel).

2.1.8 Sample Gas Manifold. A sample gas manifold is recommended for diverting portions of the sample gas stream to the analyzers. The manifold may be constructed of glass, FEP fluorocarbon, or stainless steel (type 316 or equivalent). Instead of using the manifold, separate sample lines may be connected to each analyzer.

2.1.9 Oxygen Analyzer. An oxygen analyzer is used to determine the oxygen concentration (percent O_2) of the sample gas stream.

2.1.10 Nitrogen Oxides Analyzer. A NO_x analyzer is used to determine the ppm concentration of nitrogen oxides in the sample gas stream.

2.1.11 Sulfur Dioxide Analysis. Method 6 apparatus, or equivalent, is required for sulfur dioxide determination.

2.2 Calibration Gas Specifications.

2.2.1 Zero Gas. Prepurified nitrogen is used.

2.2.2 Nitrogen Oxide Calibration Gases. Mixtures of known concentrations of NO in nitrogen are required. Nominal NO concentrations of 25, 50, and 90 percent of the instrument full scale range are needed. The 90 percent gas mixture is used to set and check the instrument span and is referred to as span gas. The 25 and 50 percent gas mixtures shall be used to validate the analyzer calibration, prior to each test.

2.2.3 Oxygen Calibration Gases. Ambient air at 20.9 percent oxygen shall be used as the span gas (high range concentration gas). A midscale calibration gas (approximately 13 percent O_2 in nitrogen) shall be used to validate the analyzer calibration prior to each test.

2.2.4 Concentration Validation. Within one month prior to test use, calibration gases shall be analyzed, by the appropriate test method specified in Section 6.2, to determine their true concentration levels. Gas concentrations that are traceable to the National Bureau of Standards and which can be demonstrated to be stable are exempted from the analysis requirements.

3. Measurement System Performance Specifications and Performance Test Procedures

3.1 Analyzer. "Span" is defined as the concentration range (specified by manufacturer) over which an analyzer will give valid readings. The spans for the analyzers used in this method shall be as follows:

3.1.1 Oxygen Analyzer: 0 to 25% O_2

3.1.2 NO_x Analyzer: 0 to 120 ppm

3.2 Analyzer Interferences and Interference Response. The "interference response" of an analyzer is defined as the output response to a component in the sample gas stream, other than the gas component being analyzed; the analyzers used in this method shall not have a total interference response of more than ±2 percent of span.

Particulate matter and water vapor are the primary interfering species for most instrumental analyzers, but these may be removed physically by using filters and condensers. Other possible specific interferents found in turbine exhaust streams include carbon monoxide, carbon dioxide, nitrogen oxides, sulfur dioxide and hydrocarbons. Each analyzing instrument may respond to one or more of these interferents in ways that alter the desired measurement.

The interference response of an analyzer is determined by measuring the total analyzer response to the gaseous components (or mixtures) listed in Table 20.1; these gases may either be introduced into the analyzer separately, or as a single gas mixture. The total interference output response of the analyzer to these components, if any, shall be determined (in concentration units). The values obtained in an interference response test shall be recorded on a form similar to Figure 20.2. If the sum of the interference responses of the test gases is greater than 2 percent of the instrument span, the analyzer shall not be used in the measurement system of this method.

An interference response test of each analyzer shall be conducted prior to its initial use in the field. Thereafter, if changes are made in the instrumentation which could alter the interference response, e.g., changes in the type of gas detector, the instruments shall be retested.

In lieu of conducting the interference response test, instrument vendor data, which demonstrate that for the test gases of Table 20.1 the interference performance specification is not exceeded, are acceptable. If these data are not available, the tests shall be made.

TABLE 20.1 INTERFERENCE TEST GAS CONCENTRATIONS

CO	500 ppm
SO_2	200 ppm
NO/NO_2	200 ppm
CO_2	10%
O_2	20.9% (Air)

FIGURE 20.2 INTERFERENCE RESPONSE

Date of Test:________________________

Analyzer Type:________________________ S/N________________

Test Gas Type	Conc.	Analyzer Output Response	% of Span

$$\% \text{ of Span} = \frac{\text{Analyzer Output Response}}{\text{Instrument Span}} \times 100$$

3.3 Analyzer Response Time. When a change in pollutant concentration occurs at the inlet of the measurement system (i.e., at probe), the change is not immediately registered by the analyzer; "response time" is defined as the amount of time that it takes for the analyzer to register a concentration value within 5 percent of the new inlet concentration. The maximum response time for the analyzers used in this method is three minutes.

To determine response time, first introduce zero gas into the system until all readings are stable; then, introduce span gas into the system. The amount of time that it takes for the analyzer to register 95 percent of the final span gas concentration is the upscale response time. Next, reintroduce zero gas into the system; the length of time that it takes for the analyzer output to come within 5 percent of the final reading is the downscale response time. The upscale and downscale response times shall each be measured three times. The readings shall be averaged, and the average upscale or downscale response time, whichever is greater, shall be reported as the "response time" for the analyzer. Response time data are recorded on a form similar to Figure 20.3. A response time test shall be conducted prior to the initial field use of the measurement system, and shall be repeated if changes are made in the measurement system.

3.4 Zero Drift. "Zero drift" is the change in analyzer output during a turbine performance test, when the input to the measurement system is a pure grade of nitrogen (zero gas). The maximum allowable

RESPONSE TIME

Date of Test____________________

Analyzer Type____________________ S/N____________________

Span Gas Concentration___________ ppm

Analyzer Span Setting____________ppm

1____________seconds

Upscale 2____________seconds

3____________seconds

Average upscale response _________seconds

1____________seconds

Downscale 2____________seconds

3____________seconds

Average downscale response_________seconds

System response time = slower average time = _________seconds.

Figure 20.3

zero drift for the analyzers used in this method is ±2 percent of the specified instrument span. The zero drift calculation is made for each gas for each turbine test run; this is done by taking the difference of the zero gas concentration values measured at the start and finish of the test (see Section 6.1). The zero drift is recorded (as a percentage of the instrument span) on a form similar to Figure 20.4.

3.5 Span Drift. "Span drift" is the change in the analyzer output during a turbine performance test, when the input to the measurement system is span gas. The maximum allowable span drift for the analyzers used in this method is ±2 percent of the specified instrument span. The span drift calculation is to be made for each gas for each turbine test run; this is done by taking the difference between the span gas concentration values measured at the beginning and end of the test. Span drift is recorded (as a percentage of instrument span) on a form similar to Figure 20.4. Span drift must be corrected for any zero drift that occurred during the test period (see Figure 20.4).

4. Procedure for Field Sampling

4.1 Selection of a Sampling Site and the Minimum Number of Traverse Points.

4.1.1 Select a sampling site as close as practical to the exhaust of the turbine. Turbine geometry, stack configuration, internal baffling, and point of introduction of dilution air will vary for different turbine designs. Thus, each of these factors must be given special consideration in order to obtain a representative sample. Whenever possible, the sampling site shall be located upstream of the

TURBINE SAMPLING SYSTEM

Zero and Span Drift Data

Turbine Type ____________________ S/N ____________________

Date: ____________________

Test No.: ____________________

Analyzer: Type ____________________ S/N ____________________

	Initial Calibration ppm or %	Final Calibration ppm or %	Difference Initial-Final ppm or %	% of Span
Zero Gas				
High Calibration Gas (Span Gas)			*	

$$\% \text{ of Span} = \frac{\text{Absolute Value of Difference}}{\text{Instrument Span}} \times 100$$

*Corrected for zero drift, i.e., if zero drift over test period is +2 ppm then 2 ppm shall be subtracted from the difference between the initial and final readings.

Figure 20.4

Reproduced from best available copy.

point of introduction of dilution air into the duct. Sample ports may be located before or after the upturn elbow, in order to accommodate the configuration of the turning vanes and baffles and to permit a complete, unobstructed traverse of the stack. The sample ports shall not be located within 5 feet or 2 diameters (whichever is less) of the gas discharge to atmosphere. For supplementary-fired, combined-cycle plants, the sampling site shall be located between the gas turbine and the boiler.

4.1.2 The minimum diameter of the sample ports shall be 3-inch nominal pipe size (NPS).

4.1.3 The minimum number of points for the preliminary O_2 sampling (Section 8.3.2) shall be as follows: (1) eight, for stacks having cross-sectional areas less than 1.5 m^2 (16.1 ft^2); (2) one sample point for each 0.2 m^2 (2.2 ft^2) of area, for stacks of 1.5 m^2 to 10.0 m^2 (16.1 - 107.6 ft^2) in cross-sectional area; and (3) one sample point for each 0.4 m^2 (4.4 ft^2) of area, for stacks greater than 10.0 m^2 (107.6 ft^2) in cross-sectional area. Note that for circular ducts, the number of sample points must be a multiple of 4, and for rectangular ducts, the number of points must be one of those listed in Table 20.2; therefore, round off the number of points (upward), when appropriate.

4.2 Cross-sectional Layout and Location of Traverse Points. After the number of traverse points for the preliminary O_2 sampling has been determined, use Method 1 to locate the traverse points.

TABLE 20.2 CROSS-SECTIONAL LAYOUT FOR RECTANGULAR STACKS

No. of traverse points	Matrix layout
9	3 x 3
12	4 x 3
16	4 x 4
20	5 x 4
25	5 x 5
30	6 x 5
36	6 x 6
42	7 x 6
49	7 x 7

4.3 Measurement System Operation.

4.3.1 Preliminaries.

4.3.1.1 Prior to the turbine test, the measurement system shall have been demonstrated to have met the performance specifications for interference response and response time described in Sections 3.2 and 3.3.

4.3.1.2 Turn on the sample pump and instruments; allow the normal warmup time required for stable instrument operation.

4.3.1.3 After the instruments have stabilized, the measurement system shall be calibrated using the procedures detailed in Section 6.1. Transfer the zero and span gas calibration data from Figure 20.5 to a form similar to Figure 20.4.

4.3.1.4 At the beginning of each NO_x test run and, as applicable, during the run, record turbine data as indicated in Figure 20.6 Also, record the location and number of the traverse points on a diagram.

4.3.2 Preliminary Oxygen Sampling.

4.3.2.1 At the start of a 3-run sample sequence, position the probe at the first traverse point and begin sampling. The minimum sampling time at each point shall be 1 minute plus the average system response time. Determine the average steady-state concentration of O_2 at each point and record the data on Figure 20.7.

4.3.2.2 Select the eight sample points at which the lowest oxygen concentrations were obtained. These same points shall be used for all three runs which comprise the emission test. More points may be used, if desired.

Figure 20.5

CALIBRATION DATA

Date________________________

Analyzer Type________________ S/N________________________

High Range Gas Conc.____________ % Full Scale________________

Mid Range Gas Conc.____________ % Full Scale________________

Low Range Gas Conc.____________ % Full Scale________________

Zero Gas____________________ % Full Scale________________

Figure 20.6

STATIONARY GAS TURBINE

TURBINE OPERATION RECORD

Test Operator____________________

Turbine ID
Type____________________
S/N____________________

Location
Plant____________________
City____________________

Ambient Temperature____________________

Ambient Humidity____________________

Test Time Start____________________

Test Time Finish____________________

Fuel Flow Rate____________________*

Water or Steam Flow Rate____________________*

Ambient Pressure ____________________

Date____________________

Ultimate Fuel Analysis
C____________________
H____________________
O____________________
N____________________
S____________________
Ash____________________
H_2O____________________

Trace Metals
____
Na____________________
Va____________________
K____________________
etc.**____________________

Operating Load____________________

---

*Describe measurement method, i.e., continuous flow meter, start finish volumes, etc.

**i.e., Additional elements added for smoke suppression.

FIGURE 20.7

Preliminary Oxygen Traverse

Location ________________ Date __________

Plant ________________________

City, State ____________________________

Turbine ID

Mfg._________________________

Model, serial number ____________________________

Sample Point	Oxygen Concentration

4.3.3 Emission Sampling.

4.3.3.1 Position the probe at the first point determined in the preceding section and begin sampling. The minimum sampling time at each point shall be 3 minutes plus the average system response time. Determine the average steady-state concentration of O_2 and NO_x at each point and record the data on Figure 20.8.

4.3.2.2 After sampling the last point, conclude the test run by recording the final turbine operating parameters and by determining the zero and span drift, as described in Sections 3.4 and 3.5. If the zero and/or span drift exceed ±2.0 percent the run may be considered invalid, or may be accepted provided the calibration data which results in the highest corrected emission concentration is used.

4.3.2.3 If additional turbine runs are conducted within 4 hours of the previous run, an initial calibration of the measurement system is not required. If more than 4 hours have elapsed between runs, the pretest calibration shall be done.

4.4 An SO_2 determination shall be made (using Method 6, or equivalent) during the test. A minimum of six total points, selected from those required for the NO_x measurement, shall be sampled; two points shall be used for each sample run. The sample time at each point shall be at least 10 minutes. The oxygen readings taken during the NO_x test runs corresponding to the SO_2 traverse points (see Section 4.3.3.1) shall be averaged, and this average oxygen concentration shall be used to correct the integrated SO_2 concentration obtained by Method 6 to 15 percent O_2 (see Equation 20-1).

Figure 20.8

STATIONARY GAS TURBINE
GAS SAMPLE POINT RECORD

Turbine ID
Mfg.____________________
Model & S/N______________
Plant____________________

Location
City____________________
State___________________

Ambient Temp.____________________
Ambient Press____________________
Ambient Humidity__________________
Date__________________________
Test Time Start__________________
Test Time Finish_________________

Test Operator Name________________________
O_2 Instrument Type__________S/N__________
NO_x Instrument Type__________S/N__________

Sample Point	Time (Min)	O_2* (%)	NO_x* (ppm)
1	0		

*Average steady state value from recorder or instrument readout.

5. Emission Calculations

5.1 Correction to 15 Percent Oxygen. Using Equation 20-1, calculate the NO_x and SO_2 concentrations (adjusted to 15 percent O_2). The correction to 15 percent oxygen is sensitive to the accuracy of the oxygen measurement. At the level of analyzer drift specified in the method (±2 percent of full scale), the change in the oxygen concentration correction can exceed 10 percent when the oxygen content of the exhaust is above 16 percent O_2. Therefore O_2 analyzer stability and careful calibration are necessary.

$$\begin{array}{c}\text{Actual Pollutant}\\ \text{Concentration}\\ (NO_x \text{ or } SO_2)\end{array} \times \frac{5.9\%}{20.9\% - O_2\%\ \text{actual}} = \begin{array}{c}\text{Pollutant concentration}\\ \text{adjusted to } 15\%\ O_2\end{array}$$

Equation 20-1

where:

5.9% is 20.9% - 15% (the defined concentration basis)

O_2 actual is the sample point oxygen concentration for NO_x calculation, and the average O_2 concentration for SO_2 calculation.

5.2 Calculate the average adjusted NO_x concentration by summing the point values and dividing by the number of sample points.

6. Calibration

6.1 Measurement System. Prior to each turbine test, the measurement system shall be calibrated according to the procedures described below. The manufacturer's operation and calibration instructions are also to be followed as required for each specific analyzer.

6.1.1 Turn on all measurement system components and allow them to warm up until stable conditions are achieved. Next, introduce zero gas and each of the calibration gases described in Section 6.2, one at a time, into the inlet of the probe. The responses of the analyzer to these gases shall be used to establish a calibration curve or to verify the manufacturers calibration curve. The data obtained in these procedures shall be recorded on a form similar to Figure 20.4. If, for the mid-scale gases, the accuracy of the manufacturer's calibration curve or the expected response curve cannot be shown to be ±2 percent of full scale (or better), the calibration shall be considered invalid and corrective measures on the instrument shall be taken. The calibration procedure shall be repeated, using only zero gas and span gas, at the conclusion of test; this allows calculation of zero and span drift (Sections 3.2 and 3.3).

6.2 Calibration Gas Mixtures.

6.2.1 Within one month prior to the turbine test, the NO_x calibration gas mixtures shall be analyzed, using the phenoldisulfonic acid procedure (Method 7) for nitrogen oxides. A minimum of three analyses shall be done, and the average concentration of each gas shall be reported as the true calibration gas value (see Figure 20.9). Alternate procedures may be employed, subject to the approval of the Administrator, to determine the calibration gas concentration.

Note: The NO_x calibration gas mixtures shall contain nitric oxide (NO) in nitrogen. Instruments which require conversion of one nitrogen

Figure 20.9

ANALYSIS OF CALIBRATION GAS MIXTURES

CYLINDER GAS COMPOSITION Reference Method Used______

Date______________________________

Low Range Calibration Gas Mixture

Sample 1____________ppm

Sample 2____________ppm

Sample 3____________ppm

Average____________ppm

Mid Range Calibration Gas Mixture

Sample 1____________ppm

Sample 2____________ppm

Sample 3____________ppm

Average____________ppm

High Range (span) Calibration Gas Mixture

Sample 1____________ppm

Sample 2____________ppm

Sample 3____________ppm

Average____________ppm

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oxide component to another for total NO_x measurement shall be checked to ensure that this conversion is complete and reproducible, as specified by the manufacturer.

6.2.2 Ambient air may be used as the oxygen span gas. The mid-scale calibration gas concentration shall be certified (by vendor) as being within ±2 percent of the indicated concentration.